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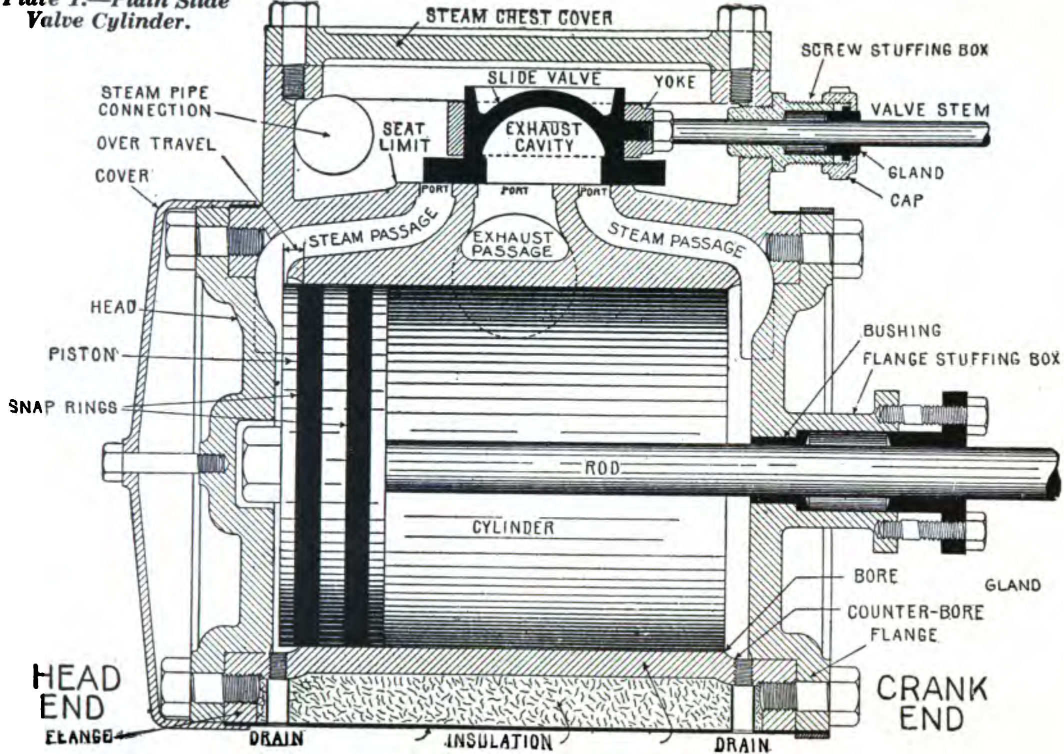
Plate 1.—Plain Slide Valve Cylinder.



Sectional view of an engine cylinder, as designed by the author, showing the various parts with their names.

AUDELS ENGINEERS AND MECHANICS GUIDE 1

A PROGRESSIVE ILLUSTRATED SERIES
WITH QUESTIONS-ANSWERS
CALCULATIONS

COVERING

MODERN ENGINEERING PRACTICE

SPECIALLY PREPARED FOR ALL ENGINEERS
ALL MECHANICS AND ALL ELECTRICIANS.
A PRACTICAL COURSE OF STUDY AND
REFERENCE FOR ALL STUDENTS AND
WORKERS IN EVERY BRANCH OF THE
ENGINEERING PROFESSION

BY
FRANK D. GRAHAM, B.S., M.S., M.E.
GRADUATE PRINCETON UNIVERSITY
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STATIONARY AND MARINE ENGINEER

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NOTE

In planning this helpful series of Educators, it has been the aim of the author and publishers to present step by step a *logical plan of study in* ***General Engineering Practice,*** taking the middle ground in making the information readily available and showing by text, illustration, question and answer, and calculation, the theories, fundamentals and modern applications, including construction *in an* **interesting** *and* **easily understandable** *form.*

Where the question and answer form is used, the plan has been to give *short, simple* and *direct answers*, limited to one paragraph, thus simplifying the more complex matter.

In order to have adequate space for the presentation of the important matter and not to divert the attention of the reader, descriptions of machines have been excluded from the main text, being printed in smaller type under the illustrations.

Leonardo Da Vinci once said:

> "Those who give themselves to ready and rapid practice before they have learned the theory, resemble sailors who go to sea in a vessel without a rudder"

—in other words, *"a little knowledge is a dangerous thing."* Accordingly the author has endeavored to give ***as much*** *information* ***as possible*** in the space allotted to each subject.

> The author is indebted to the various manufacturers for their co-operation in furnishing cuts and information relating to their products.

These books will speak for themselves and will find their place in the great field of Engineering.

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CHAPTER 1

BASIC PRINCIPLES

The Medium.—In the operation of a steam engine, **steam** is the **medium** or *working substance* by which **some** of *the heat energy, liberated from the fuel by combustion, is transmitted to the engine, and* **partly** *converted by the latter into mechanical work.*

The behavior of this medium under various conditions should therefore be thoroughly understood by the engineer. Accordingly, water, from which it is formed by the application of heat, should be first considered.

Water.—This remarkable substance is *a compound of hydrogen and oxygen in the proportion of* 2 *parts by weight of hydrogen to* 16 *parts by weight of oxygen.*

Since the atom of oxygen is believed to weigh 16 times as much as the atom of hydrogen, the molecule of water is said to contain 2 atoms of hydrogen and 1 atom of oxygen, being represented by the formula H_2O.

Under the influence of temperature and pressure this substance H_2O may exist as

1. A solid;
2. A liquid, or
3. A gas.

As a solid it is called *ice**; as a liquid, *water*, and as a gas, *steam.*

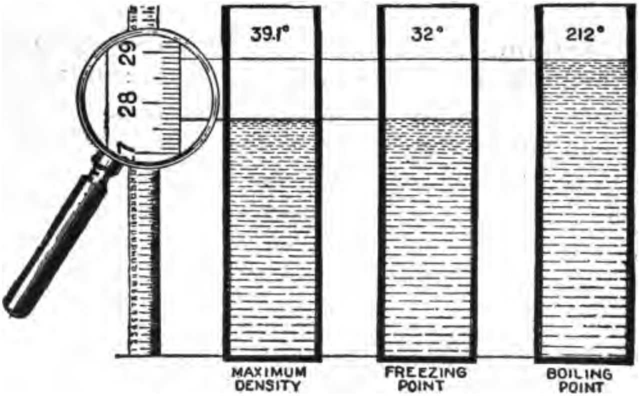
---

*NOTE.—One cu. ft. of ice at 32° Fahr., weighs 57.5 lbs.; one lb. of ice at 32° F. has a volume of .0174 cu. ft., or 30.067 cu. ins. The relative volume of ice to water at 32° F., is 1.0855, the expansion in passing into the solid state being 8.55%. Specific gravity of ice =.922, water at 62° F. being 1.—*Clark.*

Ques. What is the most remarkable characteristic of water?

Ans. Water at its maximum density (39.1 degrees F.) will expand as heat is added, and it will also expand slightly as the temperature falls from this point, as illustrated in figs. 1 to 3.

Ques. What is the freezing and boiling points of water at atmospheric pressure at the sea level?



FIGS. 1 to 3.—The most remarkable characteristic of water: *expansion below and above its temperature or "point of maximum density"* 39.1° *Fahr.* Imagine one pound of water at 39.1° F. placed in a cylinder having a cross sectional area of 1 sq. in. as in fig. 1. The water having a volume of 27.68 cu. ins., will fill the cylinder to a height of 27.68 ins. If the liquid be cooled it will expand, and at say the *freezing point* 32° F., will rise in the tube to a height of 27.7 ins., as in fig. 2, before freezing. Again, if the liquid in fig. 1 be heated, it will also expand and rise in the tube, and at say the *boiling point* (for atmospheric pressure 212° F.), will occupy the tube to a height of 28.88 cu. ins, as in fig. 3.

Ans. It will freeze at 32° Fahr., and boil at 212°, when the barometer reads 29.921 inches.*

Ques. Is the boiling point the same in all places?

Ans. No.

The boiling point of water will lower as the altitude increases; at an altitude of 5,000 feet, water will boil at a temperature of 202° Fahr.

---

*NOTE—29.921 inches of mercury = standard atmosphere = 14.696 lbs. per sq. inch.—*Marks and Davis.*

Ques. How does pressure affect the boiling point of water?

Ans. An increase of pressure will elevate the boiling point.

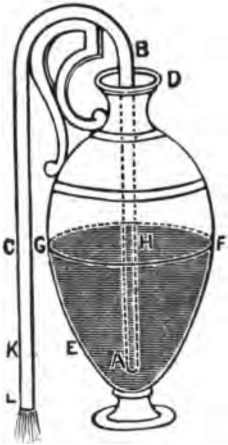


Fig. 4.—The syphon. Let A B C, be a bent syphon, or tube, of which the leg A B, is plunged into a vessel D E, containing water. If the surface of the water be F G, the leg of the syphon, A B, will be filled with water as high as the surface, that is, up to H, the portion H B C, remaining full of air. If, then the air be drawn off by suction through the aperture C, the liquid also will follow. And if the aperture C, be level with the surface of the water, the syphon, though full, will not discharge the water, but will remain full: so that, although it is contrary to nature for water to rise, it has risen so as to fill the tube A B C, and the water will remain in equilibrium, like the beams of a balance, the portion H B, being raised on high, and the portion B C, suspended. But if the outer mouth of the syphon be lower than the surface F G, as at K, the water flows out, for the liquid in K B, being heavier, overpowers and draws toward it the liquid B H. The discharge, however, continues only until the surface of the water is on a level with the mouth K, when, for the same reason as before, the efflux ceases. But if the outer mouth of the tube be lower than K, as at L, the discharge continues until the surface of the water reaches the mouth A.

Ques. What is the weight of a cubic foot of water at maximum density?

Ans. It is generally taken at the figure given by Rankine, 62.425 lbs.*

---

*NOTE.—Some authorities give as low as 62.379. The figure 62.5 commonly given is approximate. The highest authoritative figure is 62.428. At 62° Fahr., the figures range from 62.291 to 62.36. The figure 62.355 *is generally accepted as the most accurate,* though for ordinary calculations, the figure 62.4 is generally taken, this corresponding to the weight at 53° F.

Ques. What is the weight of one U. S. gallon of water?

Ans. One U. S. gallon (231 cu. ins.) of water weighs 8⅓ lbs.

The figure 8⅓ is correct when the water is at a temperature of 65° Fahr.

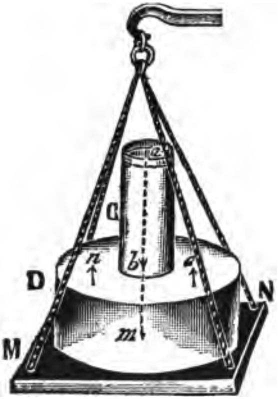


FIG. 5.—***Hydraulic principles;*** *2. Pressure exerted anywhere upon a mass of liquid is transmitted undiminished in all directions, and acts with the same force on all equal surfaces, and in a direction at right angles to those surfaces.* CD, above is a vessel composed of two cylindrical parts of unequal diameters, and filled with water to *a*. The bottom of the vessel CD, supports the same pressure as if its diameter were everywhere the same as that of its lower part; and it would at first sight seem that the scale MN, of the balance in which the vessel CD, is placed, ought to show the same weight as if there had been placed in it a cylindrical vessel having the same weight of water, and having the diameter of the part D. But the pressure exerted on the bottom of the vessel is not all transmitted to the scale MN; for the *upward* pressure upon the surface *n o*, of the vessel is precisely equal to the weight of the *extra* quantity of water which a cylindrical vessel would contain, and balances an equal portion of the *downward* pressure on *m*. Consequently the pressure on the plate MN, is simply equal to the weight of the vessel CD, and of the water which it contains.

Ques. How does the pressure of water due to its weight, vary?

Ans. It varies with the *head*, and is equal to .43302 lbs. per sq. in. for every foot of (static) head.

NOTE.—***Compressibility of water.***—Water is very slightly compressible. Its compressibility is from .00004 to .000051 for one atmosphere, decreasing with increase of temperature. For each foot of pressure, distilled water will be diminished in volume .0000015 to .0000013. Water is so incompressible that even at a depth of a mile a cubic foot of water will weigh only about ½ lb. more than at the surface.—*Kent.*

Heat.—By definition heat *is a form of energy known by its effects.*

These effects are indicated through the touch and feeling, as well as by the expansion, fusion, combustion or evaporation of the matter upon which it acts.

Ques. What is temperature?

Ans. That which indicates how hot or cold a substance is; a measure of *sensible heat.*

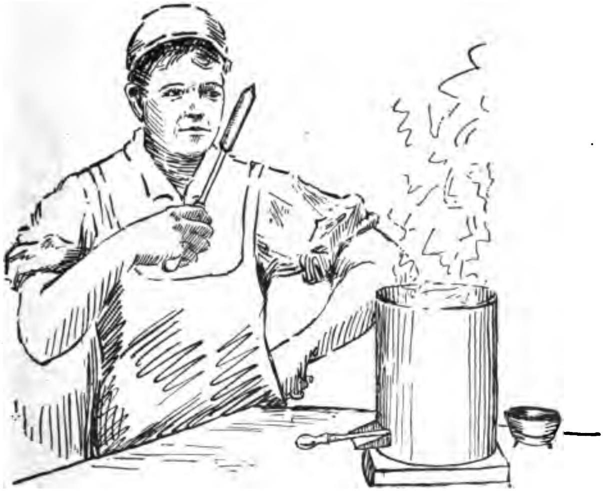


FIG. 6.—Method of judging the heat of a soldering bit or so called "iron," illustrating ***sensible heat.***

Ques. What is sensible heat?

Ans. That heat which produces a rise of temperature as distinguished from *latent heat.*

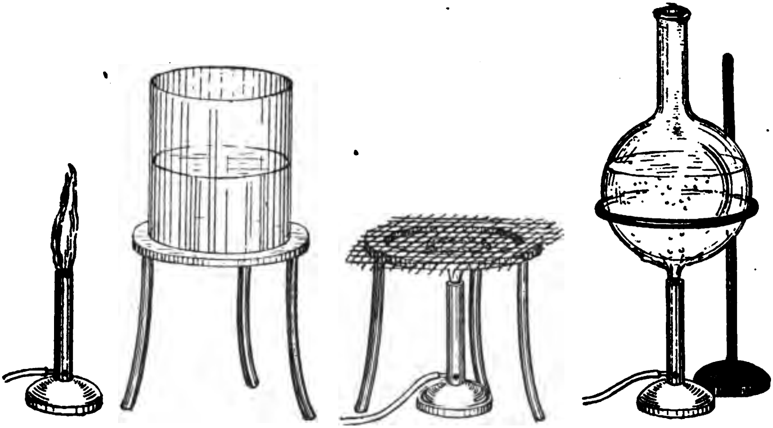
Ques. What is latent heat?

Ans. The quantity of heat required to change the *state* or condition under which a substance exists without changing its temperature.

Thus a definite quantity of heat must be transferred to ice at 32° to change it into water at the same temperature.

Ques. What is specific heat?

Ans. The ratio of the quantity of heat required to raise the temperature of a given weight of any substance one degree to the quantity of heat required to raise the temperature of the same weight of water from 62° to 63° Fahr.



FIGS. 7 to 9.—Three ways in which heat is transferred; fig. 7, by radiation; fig. 8, by conduction; fig. 9, by convection. In fig. 7, the water in the beaker is heated by *heat rays which radiate in straight lines in all directions from the flame*. In fig. 8, the flame will not pass through the wire gauze, because the latter conducts the heat away from the flame so rapidly that the gas on the other side is not raised to the temperature of ignition. In fig. 9, the water nearest the flame becomes heated and expanded. It is then rendered less dense than the surrounding water, and hence rises to the top while the colder and therefore denser water from the sides flows to the bottom thus *transferring heat by convection currents.*

Ques. Explain the term "transfer of heat."

Ans. When bodies of unequal temperatures are placed near each other, heat leaves the hot body and is absorbed by the colder body until the temperature of each is equal.

The rate by which the heat is absorbed by the colder body is proportional to the difference of temperature between the two bodies. The greater the difference of temperature, the greater the rate of flow of the heat.

Ques. How does a transfer of heat take place?

Ans. By radiation, conduction or convection.

Thus, in a boiler, heat is given off from the furnace fire in rays which radiate in straight lines in all directions being transferred to the crown and sides of the furnace by radiation; it passes through the plates by conduction, and is transferred to the water by convection, that is, by currents.

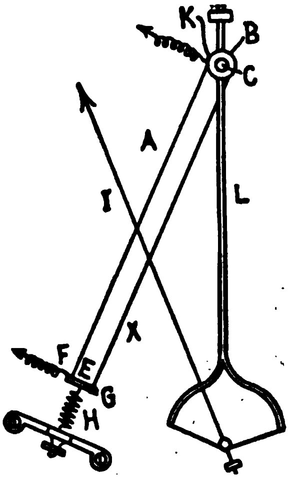


FIG. 10.—Diagram showing principle and construction of the Whitney hot wire instruments illustrating ***expansion by the action of heat.*** The action of instruments of this type depends on the heating of a wire by the passage of a current causing the wire to lengthen. This elongation is magnified by suitable mechanism and transmitted to the pointer of the instrument.

Ques. What is an important effect of heat?

Ans. Bodies expand by the action of heat. For instance, boiler plates are riveted with red hot rivets in an expanded state; on cooling the rivets contract and draw the plates together with great force making a tight joint.

An exception to the rule, it should be noted, is water, which contracts as it is heated from the freezing point 32° Fahr., to the point of maximum density 39.1°; at other temperatures it expands.

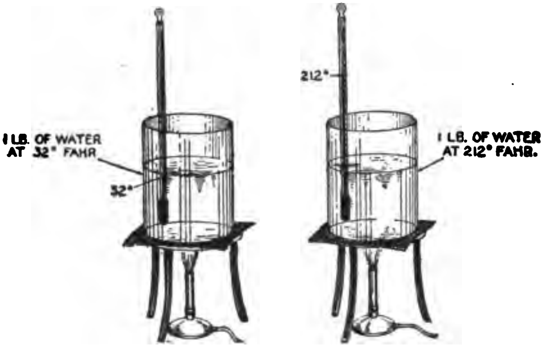
Heat and Work.—Heat develops *mechanical force,* and *motion,* hence *it is convertible into mechanical work.*

Ques. How can heat be measured?

Ans. By a standard unit called the British unit of heat, or British thermal unit (*B.t.u.*).

Ques. What is the British thermal unit?

Ans. The heat required to raise one pound of water from 62° to 63° Fahr. (*Peabody*).



Figs. 11 and 12.—Experiment illustrating the British thermal unit. Place one pound of water at 32° Fahr. into a beaker over a Bunsen burner as in fig. 11 assuming no loss of heat from the water. It will, according to the definition, require 180 heat units to heat the water from 32° to 212° Fahr. Now, if the transfer of heat take place at a uniform rate and it require, say five minutes to heat the water to 212°, then one heat unit will be transferred to the water in (5×60) ÷ 180 = 2 seconds.

Ques. What is the mean British thermal unit?

Ans. 1/180 *part of the heat required to raise the temperature of one pound of water from* 32° *to* 212° *Fahr.** †

---

*NOTE.—The old definition of the heat-unit (*Rankine*), viz., the quantity of heat required to raise the temperature of 1 lb. of water 1° Fahr., at or near its temperature of maximum density (39.1° F.) was the standard till 1909.

†NOTE.—By Peabody's definition, the heat required to raise 1 lb. of water from 32° to 212° is 180.3 instead of 180 units, and the latent heat at 212° is 969.7 instead of 970.4.

It should be noted that this is the definition adopted in this work for the British thermal unit (B. t. u.), corresponding to the unit used in the Marks and Davis steam tables, which is now the recognized standard.

Ques. What is work?

Ans. *The overcoming of resistance through a certain distance by the expenditure of* **energy.**

Ques. How can work be measured?

Ans. By a standard unit called the *foot pound.*

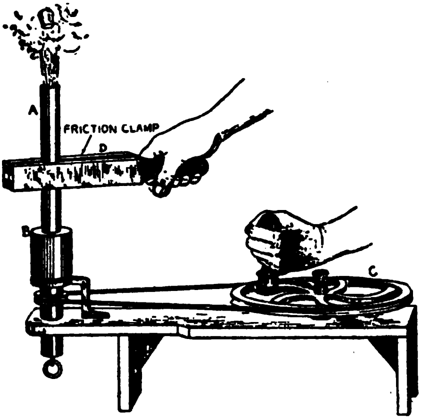


Fig. 13.—Experiment showing relation between heat and work. Take a brass tube A B, attached to a spindle geared to rotate rapidly and partly fill the tube with water and insert a cork. Apply a friction clamp D, and rapidly rotate the tube by turning the wheel C. The energy expended in overcoming the friction due to the clamp and rotating the tube causes the water to heat and finally boil; if continued long enough, the pressure generated will expel the cork. During the operation work *has been transformed into* heat.

Ques. What is a foot pound?

Ans. It is *the amount of work done in raising one pound one foot*, or in overcoming a pressure of one pound through a distance of one foot.

Thus, if a 5 pound weight be raised 10 feet, the work done is 5×10=50 foot pounds.

Ques. What is the relation between the unit of heat and the unit of work?

Ans. It was shown by experiments made by Joule (1843–50) that 1 *unit of heat*=772 *units of work*. This is known as the "mechanical equivalent of heat" or Joule's equivalent.

More recent experiments by Prof. Rowland (1880) and others give higher figures; 778 is generally accepted, but 777.5 is probably more nearly correct, the value 777.52 being used by Marks and Davis in their steam tables.

The value 778 is sufficiently accurate for ordinary calculations.

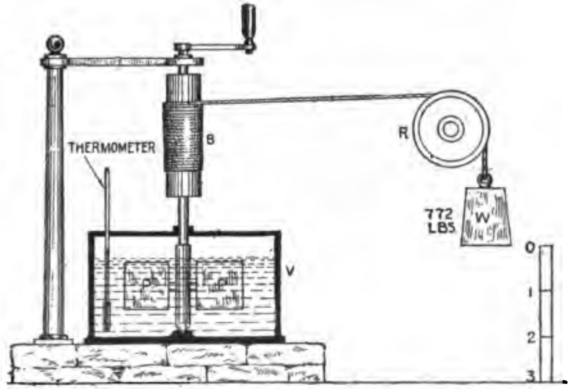
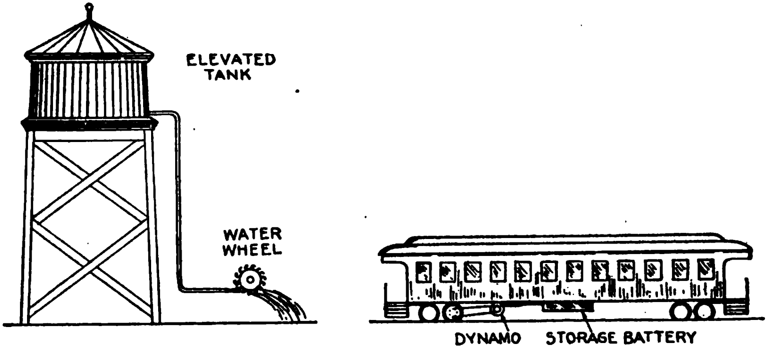


Fig. 14.—The mechanical equivalent of heat. In 1843, Dr. Joule of Manchester, England, performed his classic experiment, which revealed to the world the mechanical equivalent of heat. As shown in the figure, a paddle was made to revolve with as little friction as possible in a vessel containing a pound of water whose temperature was known. The paddle was actuated by a known weight falling through a known distance. *A pound falling through a distance of one foot represents a foot pound of work.* At the beginning of the experiment a thermometer was placed in the water, and the temperature noted. The paddle was made to revolve by the falling weight. When 772 foot pounds of energy had been expended on the pound of water, the temperature of the latter had risen one degree, and the relationship between heat and mechanical work was found; the value 772 foot pounds is known as Joule's equivalent. More recent experiments give higher figures, the value 778, is now generally used but according to Kent 777.62 is probably more nearly correct. Marks and Davis in their steam tables have used the figure 777.52.

Joule's Experiment.—In fig. 14, a weight W is attached to a cord which passes over a pulley R, and is wound around a revolving drum B. Attached to the drum is a spindle having fastened at its lower end vanes or paddles P P made of thin pieces of sheet metal. These paddles are immersed in a vessel V, containing a definite quantity of water.

In operation, as the weight W, falls, the paddles rotate in the water, the water itself being kept from rotating by fixed pieces not shown. It was discovered that the work done by the weight in descending, was not lost but appeared as heat in the water, the agitation of the paddles having increased the temperature of the water by an amount which can be measured by a thermometer.



Figs. 15 and 16.—Potential, and kinetic energy. In fig. 15, the water stored in the elevated tank possesses energy by virtue of its position; being higher than the water wheel, the water will flow by gravity through the pipe and do work on the wheel. Thus, the potential energy of the water at rest in the tank, is, when it flows through the pipe converted into kinetic energy which is spent on the wheel. Fig 16 represents a railway car with axle lighting system. If the car be set in motion and then no further power be applied its momentum or kinetic energy will drive the dynamo which in turn will charge the storage battery, and acting like a brake will gradually bring the car to rest. During this operation, the kinetic energy, originally possessed by the moving car, is absorbed by the dynamo (neglecting friction) and delivered to the battery as electrical energy which may be used in lighting the car.

By numerous experiments, Joule determined with the utmost care that one pound of water was increased in temperature one degree by the work done on it during the descent of 772 pounds through one foot.

This value, as before mentioned, is too small for ordinary calculation, the value 778, the generally accepted standard, should be used; the value 777.52 is probably more nearly correct.

Energy.—By definition, *energy is stored work*, that is, the ability to do work, or in other words, to move against resistance.

A body may possess energy whether it do any work or not, but no work is ever done except by the expenditure of energy. There are two kinds of energy:

1. *Potential energy;*
2. *Kinetic energy.*

Potential energy is *energy due to position*, as represented, for instance, by a body of water stored in an elevated reservoir, capable of doing work by means of a water wheel.

Kinetic energy is *energy due to momentum*, that is to say, the energy of a moving body.

Conservation of Energy.—The doctrine of physics, that energy can be transmitted from one body to another or transformed in its manifestations, but *may neither be created nor destroyed.*

Energy may be dissipated, that is, converted into a form from which it cannot be recovered, as is the case with the great percentage of heat escaping with the exhaust of a locomotive, or the condensing water of a steamship, but *the total amount of energy in the universe*, it is argued, *remains constant and invariable.*

Thus, in Joule's experiment, fig. 14, the potential energy of the weights is not lost, but is transformed into heat, which is one form of energy. The apparatus possesses the same amount of energy at the end as at the beginning of the experiment, but the distribution of the energy has been changed, that is the potential energy given up by the weight has been transformed into heat, raising the temperature of the water.

Power.—By definition, power is *the* **rate** *at which work is done;* in other words, it is *work divided by the* **time** *in which it is done.*

The unit of power in general use is the ***horse power**** which is defined as *33,000 foot pounds per minute.*

That is, one horse power* is required to raise a weight of
33,000 pounds 1 foot in one minute
3,300 pounds 10 feet in one minute
33 pounds 1,000 feet in one minute
3.3 pounds 10,000 feet in one minute
1 pound 33,000 feet in one minute
etc.

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*NOTE.—The term "horse power" is due to James Watt, who figured it to represent the power of a strong London draught horse to do work during a short interval, and used it as a power rating for his engines

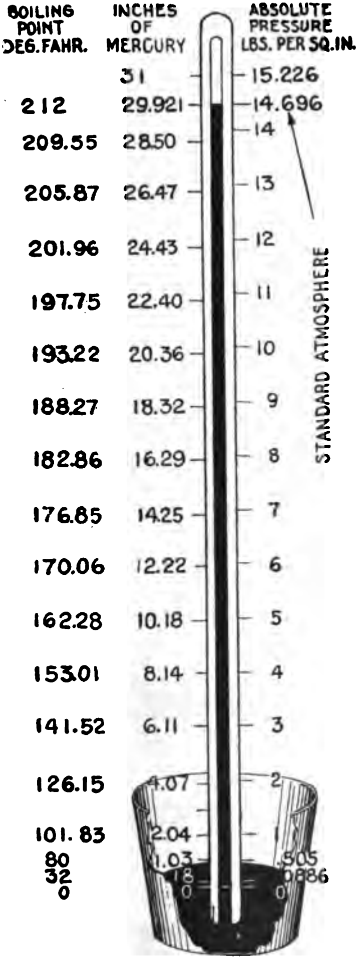


FIG. 17.—Mercurial barometer, illustrating the boiling point for various pressures.

Pressure.—According to Rankine, the term *pressure*, in the popular sense, which is also the sense generally employed in applied mechanics, is used to denote *a force, of the nature of a thrust, distributed over a surface;* in other words, the kind of force with which a body tends to expand, or resists an effort to compress it.

In the definition it should be carefully noted that the pressure is considered as *distributed over a surface.*

The pressure distributed over a surface is usually stated in terms of the pressure distributed over a unit area of the surface, as *pounds per square inch*, meaning that a pressure of a given number of pounds is distributed over each square inch of surface. This should be very clearly understood by the engineer, as further explained in figs. 18 and 19.

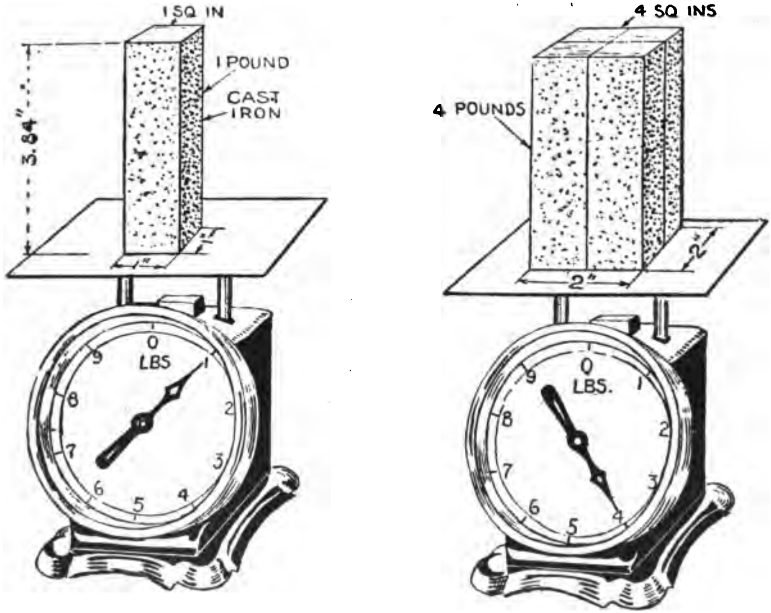
NOTE.—***Boiling point*** of various substances at atmospheric pressure (14.7 lbs.):

Ether, sulphide	100°	Fahr.
Carbon bisulphide	118°	"
Chloroform	140°	"
Bromine	145°	"
Wood spirit	150°	"
Alcohol	173°	"
Benzine	176°	"
Water	212°	"
Average sea water	213.2°	"
Saturated brine	226°	"
Nitric acid	248°	"
Oil of turpentine	315°	"
Aniline	363°	"
Naphthaline	428°	"
Phosphorus	554°	"
Sulphuric acid	590°	"
Linseed oil	597°	"
Mercury	676°	"
Sulphur	800°	"

Ques. What is atmospheric pressure?

Ans. It is the force exerted by the weight of the atmosphere on every point with which it is in contact.

At sea level this pressure is for ordinary calculations taken at 14.7 pounds per square inch, and roughly, 15 lbs. per square inch.



FIGS. 18 and 19.—*Pressure per square inch.* Cast iron weighs .26 lbs. per cu. in.; hence, a piece 1 ÷ .26 = 3.84 ins. long will weigh one pound. Place this on a scale as in fig. 18, and the pointer will register one pound. There is then in this case a pressure of one pound *distributed over a surface of* 1″×1″ = 1 sq. in. Now take 4 pieces of iron as in fig. 19 and place them on the scale as in fig. 19. The pointer will register 4 lbs., indicating a total pressure of 4 lbs. on the scale, but since this pressure is distributed over an area of 2″×2″ = 4 sq. ins., it represents a pressure of 4 lbs. ÷ 4 sq. ins. = 1 *pound per square inch.*

Ques. Why do we not feel this pressure?

Ans. Because air presses the body both externally and internally so that the pressures in different directions balance.

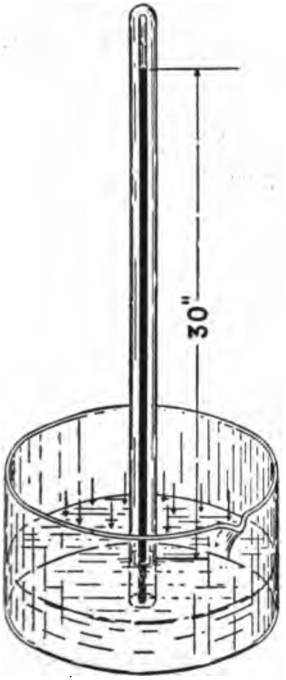


FIG. 20.—The barometer, an instrument for measuring the pressure of the atmosphere. A glass tube closed at one end is filled with mercury, inverted, and the open end submerged in a mercury well. The mercury in the tube will stand at a certain height depending on the pressure of the atmosphere.

Ques. Does the pressure of the atmosphere remain constant in any one place?

Ans. No. It continually varies depending on the conditions of the weather.

Ques. Is the pressure of the atmosphere the same in different places?

Ans. It varies with the elevation. The pressure decreases approximately one-half pound for every 1,000 feet of ascent.

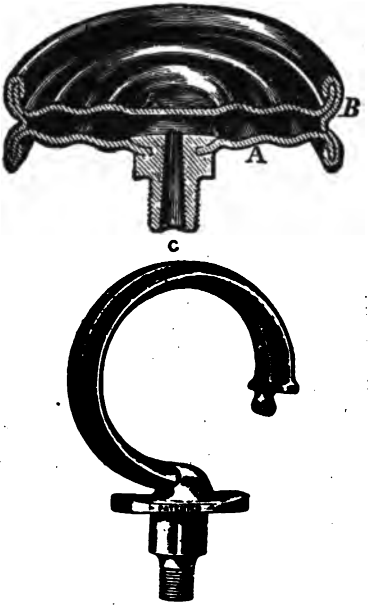
Ques. How is the pressure of the atmosphere measured?

Ans. By an instrument called the *barometer*.

Ques. Explain the principle of the barometer.

Ans. A glass tube 33 to 34 inches long is sealed at one end and filled with pure mercury, then inverted in an open cup of mercury as shown in fig. 20. The mercury will fall in the tube until its height above the mercury in the cup is about 30 inches. It remains suspended at this height because the weight of a column of mercury 30 inches high is the same as the weight of a like column of air about 50 miles high.

Pressure Scales.—The term vacuum is a much abused word; strictly speaking it is defined as *a space devoid of matter.* This is equivalent to saying *a space in which the pressure is zero.*



FIGS. 21 and 22. Bent tube and diaphragm of corrugated metal as used in two types of steam gauge. In the one class, the pressure of the steam acts upon diaphragms or plates of some kind, shown in fig. 21, which represents a section of a pair of metal plates, A A, of this kind. These are made with circular corrugations, as shown in section and also by the shading. The steam enters by the pipe *c*, and fills the chamber between the metal plates or diaphragms. The corrugations of the latter give them sufficient elasticity, so that when the pressure is exerted between them they will be pressed apart by the steam. If they were flat, it is plain that they would not yield, or only to a very slight degree, to the pressure of the steam. In the other class of gauge, the steam acts upon a bent metal tube of a flattened or elliptical section, such as shown in fig. 22. The pressure has a tendency to straighten this tube, and this straightening tendency is directly proportioned to the pressure; the free end of this tube is connected through suitable gearing to the pointer or hand.

The word vacuum has come, by ill usage to mean *any space in which the pressure is less than that of the atmosphere,* and accordingly, it is necessary to accept the latter definition.

This gives rise to two scales of pressure:

1. Gauge pressure;
2. Absolute pressure.

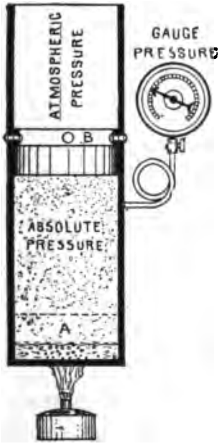


Fig. 23.—Elementary boiler or closed vessel illustrating the difference between *gauge*, and *absolute pressure*.

When the hand of a steam gauge is at zero, the pressure actually existing is 14.73 lbs. (referred to a 30 inch barometer) or that of the atmosphere. The scale in the gauge is not marked at this point 14.73 lbs. but zero because in the steam boiler as well as any other vessel under pressure, the important measurement is the difference of pressure between the inside and outside. This difference of pressure or effective pressure for doing work is called the "gauge pressure" because it is measured by the gauge on the boiler. The second pressure scale is known as ***absolute pressure***, because it gives the actual ***pressure above zero***. In all calculations relative to the expansion of steam the absolute pressure scale must be used.

Ques. How is gauge pressure expressed as absolute pressure?

Ans. By adding 14.73, or for ordinary calculations, 14.7 lbs.

Thus 80 lbs. gauge pressure$=80+14.7=94.7$ lbs. absolute pressure.

Ques. How is absolute pressure expressed as gauge pressure.

Ans. By subtracting 14.7.

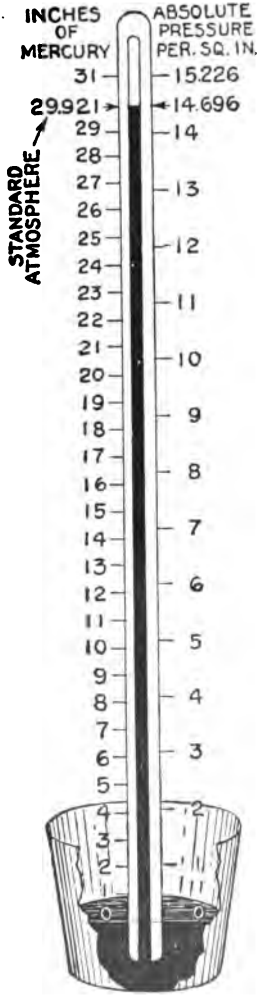


FIG. 24.—Mercurial barometer illustrating the relation between "inches of mercury" and absolute pressure in lbs. per sq. in.

Thus 90 lbs. absolute pressure = 90 — 14.7 = 75.3 lbs. gauge pressure.

Ques. How are pressures below that of the atmosphere usually expressed?

Ans. As pounds per square inch in making calculations, or the equivalent in "inches of mercury" in practice.

Thus, in the engine room, the expression "28 inch vacuum" would signify an absolute pressure in the condenser of .946 lb. per sq. in. absolute, that is to say, the mercury in a mercury column connected to a condenser having a 28 inch vacuum, would rise to a height of 28 inches, representing the difference between the pressure of the atmosphere and the pressure in the condenser, or

$$14.73 - .946 = 13.784 \text{ lbs.}$$

referred to a 30 inch barometer.

Ques. What is the meaning of the term "referred to a 30 inch barometer?"

Ans. It means that the variable pressure of the atmosphere is in value such that it will cause the mercury in the barometer to rise 30 inches.

Ques. How is the pressure in pounds per square inch obtained from the barometer reading?

Ans. Barometer reading in inches × .49116 = pressure per sq. inch.

Thus, a 30 inch barometer reading signifies a pressure of

.49116×30 = 14.74 lbs. per sq. in.

The following table gives the pressure of the atmosphere in pounds per square inch for various readings of the barometer.

Pressure of the atmosphere per square inch for various readings of the barometer:

Rule.—*Barometer in inches of mercury*×.49116 = *lbs. per sq. in.*

Barometer (ins. of mercury)	Pressure per sq. ins., lbs.	Barometer (ins. of mercury)	Pressure per sq. ins., lbs.
28.00	13.75	**29.921**	**14.696**
28.25	13.88	30.00	14.74
28.50	14.00	30.25	14.86
28.75	14.12	30.50	14.98
29.00	14.24	30.75	15.10
29.25	14.37	31.00	15.23
29.50	14.49		
29.75	14.61		

The above table is based on the standard atmosphere, which by definition = **29.921** ins. of mercury = **14.696** lbs. per sq. in., that is 1 in. of mercury = 14.696 ÷ 29.921 = **.49116** lbs. per sq. in.

Temperature Scales.

Temperature Scales.—Temperature is a measure of *sensible heat*, that is, the temperature of a substance indicates how hot or cold it is.

The instrument for measuring temperature is the well known thermometer. Briefly, it consists of a hollow stem or tube of glass with an enlargement or bulb at the foot filled with mercury which expands into the tube. The stem being uniform in bore, and the apparent expansion of mercury in the tube being equal for equal increments of temperature, it follows that if the scale be graduated with equal intervals, these will indicate equal increments or "degrees" of temperature.

There are three kinds of thermometer scales in general use:

1. Fahrenheit;
2. Centigrade;
3. Reaumur.

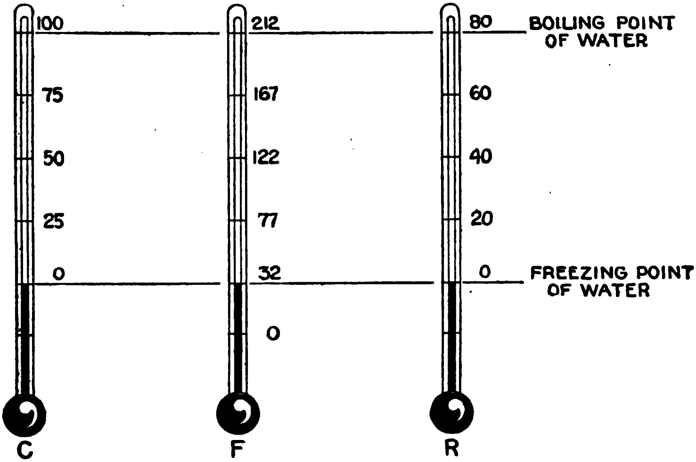
The relation between these scales is shown in figs. 25 to 27.

The Fahrenheit scale is generally used in English speaking countries, the freezing point is 32°, and boiling point, 212°.

The Centigrade scale is used in France. The freezing point is 0°, and boiling point, 100°.

The Reaumur scale is used in Russia, Sweden, Turkey and Egypt. The freezing point is zero and boiling point 80°.

Fahrenheit is converted into Reaumur by deducting 32° and taking four-ninths of the remainder, and Reaumur into Fahrenheit by multiplying by nine-fourths, and adding 32° to the product.



Figs. 25 to 27.—Various thermometer scales. Fig 25, Centigrade; fig. 26, Fahrenheit; fig. 27, Reaumur. From the figure the scales may be clearly compared, and degrees converted from one scale to another without calculation.

Centigrade temperatures are converted into Fahrenheit temperatures by multiplying the former by 9, dividing by 5, and adding 32° to the quotient; and conversely, Fahrenheit temperatures are converted into Centigrade by deducting 32° and taking 5-9ths of the remainder.

Reaumur degrees are multiplied by five-fourths to convert them into the equivalent Centigrade degrees; conversely, four-fifths of the number of Centigrade degrees give their equivalent in Reaumur degrees.

Steam.—It has been stated that ***steam*** is the ***medium*** or *working substance by which* ***some*** *of the heat energy, liberated from the fuel by combustion is transmitted to the engine and* ***partly*** *converted into mechanical work.*

Owing to the peculiarities of steam, it is important that engine operators should know its exact nature or behavior under different conditions.

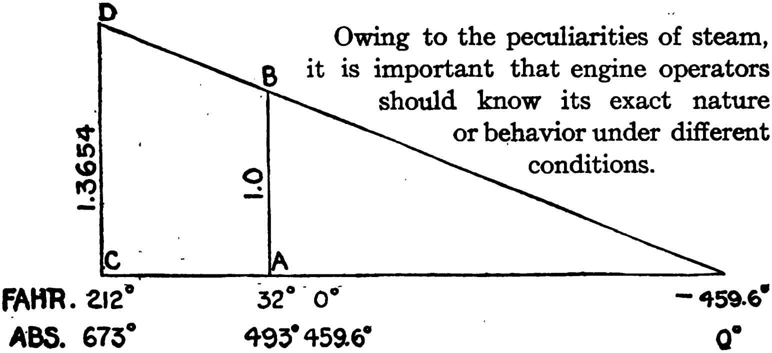


FIG. 28.—Graphical method of determining the absolute zero. It is found by experiment that when air is heated or cooled under constant pressure, its volume increases or decreases in such a way that if the volume of the gas at freezing point of water be 1 cu. ft., then its volume when heated to the boiling point of water, will have expanded to 1.3654 cu. ft. Or, inversely, if the volume remain constant, and the pressure exerted by the gas at freezing point = 1 atmosphere, then the pressure at boiling point of water = 1.3654 atmospheres. These results may be set out in the form of a diagram, as here shown. ***In construction,*** draw a vertical line to represent temperatures to any scale, and mark on it points representing the freezing point and boiling point of water, marked 32° and 212° respectively. From 32° set out, at right angles to the line of temperature, a line of pressure AB = 1 atmosphere to any scale, and at 212° a line CD = 1.3654 atmospheres to the same scale. Join the extremities DB, of these lines to intersect the line of temperatures, It is assumed by physicists that, since the pressures vary regularly per degree of change of temperature between certain limits within the range of experiment, they vary also at the same rate beyond that range, and, therefore, that the point of intersection of the straight line DB, produced gives the point at which the pressure is reduced to zero, this point being known as the absolute zero.

NOTE.—***Absolute temperature.***—This is defined as ***the actual temperature of anything reckoned from absolute zero.*** It is taken as the temperature indicated by the thermometer or similar instrument, to which is added 273.1° centigrade or 459.6° Fahrenheit, the difference between absolute zero and the zeros of the respective thermometric scales, which are arbitrarily fixed.

NOTE.—***Absolute zero.***—In physics, temperature or the heat which it represents is regarded as a manifestation of molecular activity in any substance, the higher the temperature, the greater the motion or vibration among the molecules of which every solid, liquid or gaseous body is composed. Experiments have demonstrated that a gas expands when at the freezing point and under constant pressure about $\frac{1}{491.6}$ of its volume for each increase of 1° Fahr. in pressure. This tends to show, that at some point about 491.6°—32° or 459.6° below zero or Fahrenheit's scale, the volume of the gas would have become zero or it would have lost all the molecular vibration which manifests itself as heat. The temperature of this ***absolute zero point,*** from which all temperatures of gases are reckoned, is estimated at—273.1° C. or —459.6° F. The lowest temperatures yet obtained by anyone are those at which ***hydrogen*** liquifies, —423° F., and its freezing point, 430.6° F.

Ques. What is steam?

Ans. Steam is the vapor of water.

It is a colorless, expansive, *invisible* fluid. The white cloud seen issuing from an exhaust pipe, and usually called steam, is not steam but in reality a fog of minute liquid particles produced by *condensation*.

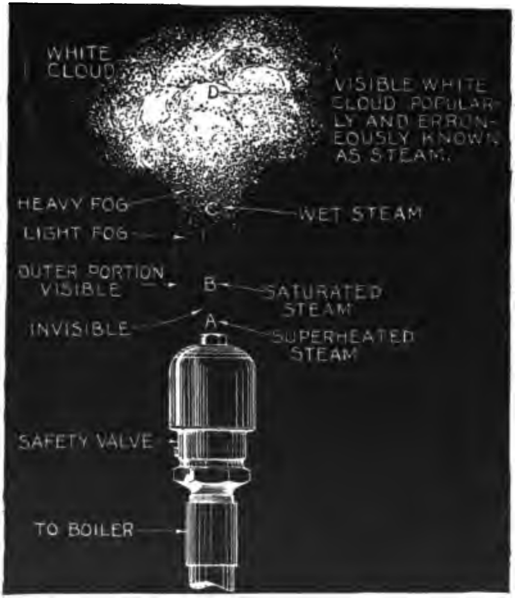


Fig. 29.—The various states of steam as exemplified in the operation of a safety valve. By closely observing a safety valve when blowing off, as for instance the safety valve on a locomotive, or better the safety valve on a marine boiler, furnishing superheated steam, very interesting phenomena can be observed. At A, very close, the escaping gas is entirely invisible being at this point superheated. At B, the outline of the ascending column is seen, the interior being invisible and gradually becoming "foggy" and as the vapor ascends from B to D, denoting the gradual reduction in temperature, the steam becoming saturated and superheated or wet, reaching the white state at D, where it is popularly and erroneously known as "steam." ***Steam is invisible.*** The reason the so called wet steam can be seen is because wet steam is a mechanical mixture made up of saturated steam which is invisible, and which holds in suspension a multiplicity of fine water globules formed by condensation; it is the collection of water globules or *condensate* that is visible.

Ques. How is steam classified according to its quality?

Ans. As *wet*, *dry*, *saturated*, *superheated*, or *gaseous*.

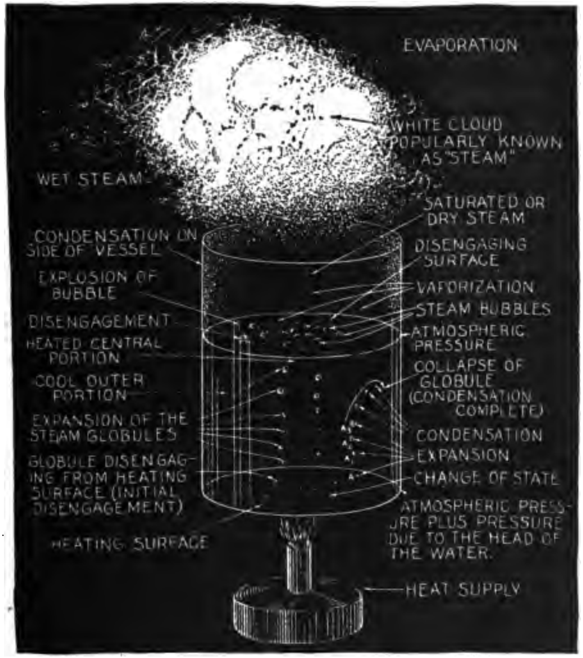

EVAPORATION
WHITE CLOUD POPULARLY KNOWN AS "STEAM"
WET STEAM
SATURATED OR DRY STEAM
CONDENSATION ON SIDE OF VESSEL
DISENGAGING SURFACE
EXPLOSION OF BUBBLE
VAPORIZATION
STEAM BUBBLES
DISENGAGEMENT
ATMOSPHERIC PRESSURE
HEATED CENTRAL PORTION
COLLAPSE OF GLOBULE (CONDENSATION COMPLETE)
COOL OUTER PORTION
EXPANSION OF THE STEAM GLOBULES
CONDENSATION
EXPANSION
GLOBULE DISENGAGING FROM HEATING SURFACE (INITIAL DISENGAGEMENT)
CHANGE OF STATE
ATMOSPHERIC PRESSURE PLUS PRESSURE DUE TO THE HEAD OF THE WATER.
HEATING SURFACE
HEAT SUPPLY

Fig. 30.—The phenomena of *vaporization*. When heat is applied to water in a vessel as shown, it is conducted through the heating surface to the lower state which gradually becomes heated to the boiling point. This is followed by the formation of globules of steam on the heating surface indicating that particles of the water have received a supply of heat equal to the sensible and latent heat of steam at the pressure existing at the bottom of the vessel, thus a *change of state* has taken place, and this may be called *initial vaporization* as distinguished from vaporization or the completion of the process. As more heat is added, more of the water adjacent to the globules is converted into steam which causes the globules to increase in size until their buoyancy becomes sufficient to overcome the tension with the heating surface and *initial disengagement* takes place. Following the course of a globule disengaging from the central and hottest portion of the heating surface, it rapidly rises to the surface, and expands as it rises because the pressure gradually decreases due to diminishing head of water. On reaching the *disengaging surface*, a bubble is formed which at once bursts as the water closes in behind the steam contained in the bubble, thus completing the process of vaporization of the original particles of water; that is to say, a *change of state* has taken place and the steam has been disengaged from the water.

NOTE.—It should be noted in the above illustration that all of the steam globules formed on the heating surface do not reach the surface, as for instance, the globule A, found near the side of the vessel will, as it rises, take same course as A1, A2, A3, expanding as it rises. After passing the portion A3, it may be deflected over toward the side of the vessel into relatively cold water as indicated by the arrow, giving up its heat to the cold water, resulting in *condensation* and the gradual collapse of the globule as indicated. It should be noted further that the pressure at the bottom of the vessel or heating surface being greater than the pressure at the disengaging surface, the temperature at which initial vaporization takes place is greater than that of vaporization proper.

Wet steam contains intermingled moisture, mist or spray, and has the same temperature as dry saturated steam of the same pressure.

Dry steam contains no moisture; it may be either saturated or superheated.

Saturated steam is steam of a temperature due to its pressure.

**Superheated steam* is steam having a temperature *above* that due to its pressure.

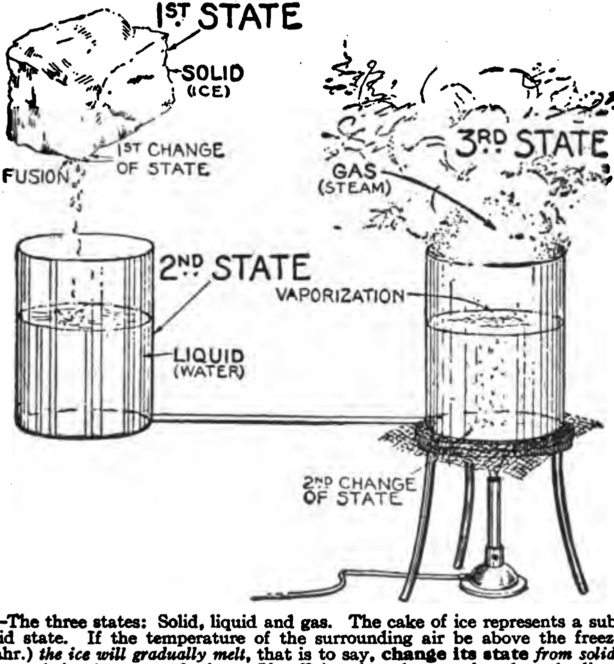


Fig. 31.—The three states: Solid, liquid and gas. The cake of ice represents a substance in the solid state. If the temperature of the surrounding air be above the freezing point (32° Fahr.) *the ice will gradually melt*, that is to say, change its state *from solid to liquid* this process being known as fusion. If sufficient heat be transferred to the liquid, *it will boil*, that is to say, change its state *from liquid to gas*, this process being known as vaporization. Very interesting phenomena take place during these changes, which are explained in the accompanying text.

Ques. Under what conditions does steam exist?

Ans. When there is the proper relation between the temperature of the water and the external pressure. For instance, for a given temperature of the water there is a certain external pressure above which steam will not form.

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*NOTE.—A term sometimes, though ill advisedly, used for highly superheated steam is gaseous steam or steam gas. The saving in the water consumption of a steam engine due to superheating the steam is a little over one per cent for each ten degrees of superheat.

Ques. How is steam produced?

Ans. By heating water until it reaches the *boiling point.*

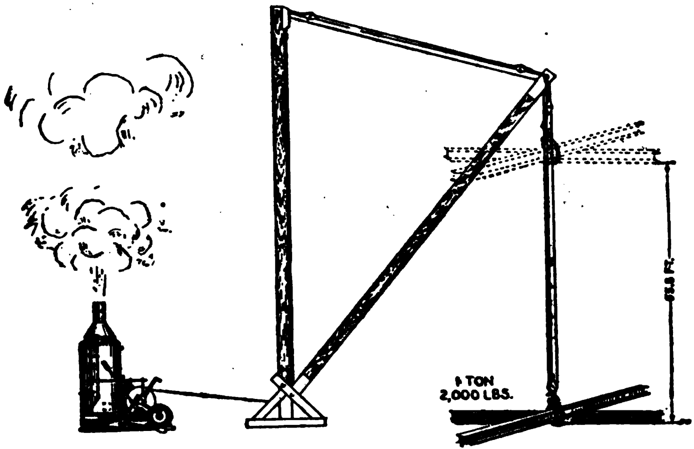
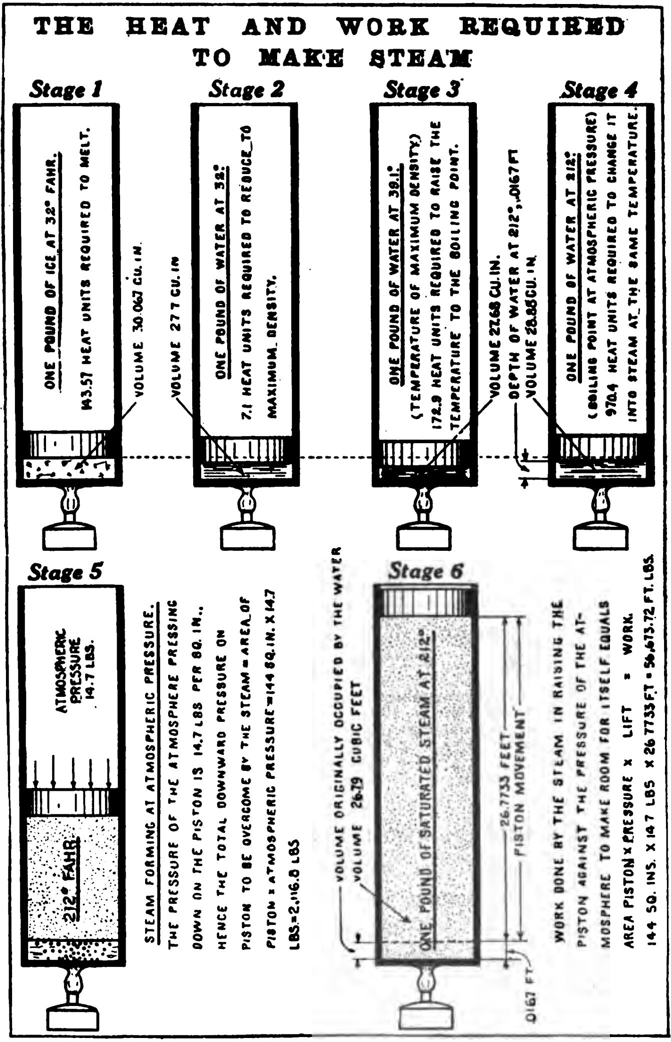

1 TON
2,000 LBS.
55.8 FT.

FIG. 32.—The fusion of ice, illustrating the ***work done when the pound of ice at 32° Fahr. is melted or converted into water at the same temperature.*** The latent heat of fusion being 143.57 heat units, and since one heat unit is equivalent to 778 ft. lbs. the work done during the fusion of one pound of ice is 778×143.57 = 111,698 ft. lbs. This is approximately equivalent to the work done when a hoisting engine hoists 2,000 lbs. a distance of 55.8 ft. as shown in the illustration.

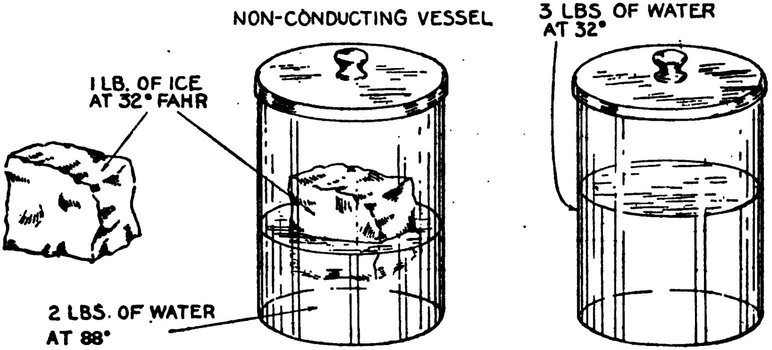
What an Engineer should know about Steam.—As has been stated, the *medium* which transmits heat energy to the engine to be transformed into mechanical work is a very remarkable substance; it disobeys the general law of expansion by heating, and, can exist in three different states, that is, as

1. A solid (ice);
2. A liquid (water), or
3. A gas (steam),


THE HEAT AND WORK REQUIRED TO MAKE STEAM
Stage 1
ONE POUND OF ICE AT 32° FAHR.
143.57 HEAT UNITS REQUIRED TO MELT.
VOLUME 30.067 CU. IN.
Stage 2
VOLUME 27.7 CU. IN
ONE POUND OF WATER AT 32°
7.1 HEAT UNITS REQUIRED TO REDUCE TO MAXIMUM DENSITY.
Stage 3
ONE POUND OF WATER AT 39.1°
(TEMPERATURE OF MAXIMUM DENSITY)
172.9 HEAT UNITS REQUIRED TO RAISE THE TEMPERATURE TO THE BOILING POINT.
VOLUME 27.68 CU. IN.
DEPTH OF WATER AT 212°, .0167 FT
VOLUME 28.85 CU. IN.
Stage 4
ONE POUND OF WATER AT 212°
(BOILING POINT AT ATMOSPHERIC PRESSURE)
970.4 HEAT UNITS REQUIRED TO CHANGE IT INTO STEAM AT THE SAME TEMPERATURE.
Stage 5
ATMOSPHERIC PRESSURE 14.7 LBS.
212° FAHR.
STEAM FORMING AT ATMOSPHERIC PRESSURE.
THE PRESSURE OF THE ATMOSPHERE PRESSING DOWN ON THE PISTON IS 14.7 LBS PER SQ. IN., HENCE THE TOTAL DOWNWARD PRESSURE ON PISTON TO BE OVERCOME BY THE STEAM = AREA OF PISTON × ATMOSPHERIC PRESSURE = 144 SQ. IN. × 14.7 LBS. = 2,116.8 LBS
Stage 6
VOLUME ORIGINALLY OCCUPIED BY THE WATER
VOLUME 26.79 CUBIC FEET
ONE POUND OF SATURATED STEAM AT 212°
26.7733 FEET
PISTON MOVEMENT
.0167 FT.
WORK DONE BY THE STEAM IN RAISING THE PISTON AGAINST THE PRESSURE OF THE ATMOSPHERE TO MAKE ROOM FOR ITSELF EQUALS
AREA PISTON × PRESSURE × LIFT = WORK.
144 SQ. INS. × 14.7 LBS × 26.7733 FT = 56,673.72 FT. LBS.

FIGS. 33 to 38.—***From ice to steam,*** illustrating the *six stages* in the making of steam from ice at 32° Fahr.

depending upon conditions of pressure and temperature. These changes are shown in the accompanying series of diagrams, (figs. 33 to 38), which represent the several stages in transforming a pound of ice at 32° Fahr. into saturated steam at 212°, the temperature corresponding to atmospheric pressure. The engineer should make a careful study of these diagrams and the matter following to properly understand the *nature* of the medium he has to deal with in the operation of an engine.



FIGS. 39 to 41.—Experiment illustrating the ***latent heat of fusion.*** ***It requires*** *144 heat units to "melt" a pound of ice at 32° Fahr., that is, to convert it into water of the same temperature. Accordingly, if a pound of ice (fig. 39) be placed in a non-conducting vessel with two pounds of water at 88° Fahr., it will be found that when all of the ice has been melted by the transfer of heat from the water to the ice the temperature of the mixture (fig. 41) of melted ice and the water will be the same as the original temperature of the ice, 32°. ***The reason*** for this is because the total heat above 32° in the water at 88° was the same as the latent heat of the ice, or 144 heat units, that is to say, the total heat above 32° in the water was (88 × 2) — 32 = 144 heat units. ***It should be understood*** that the term non-conducting vessel implies one in which allows no heat to pass through its sides. Such a vessel is not possible to construct, but by covering an ordinary vessel all over with a thick layer of asbestos very little heat will be lost.

It will be noted from the diagrams that the stages into which the process of transforming ice into steam by heating are

1. Fusion of the ice (at 32° F.);
2. Contraction of the water (between 32° and 39.1°);
3. Expansion of the water (between 39.1° and 212°);

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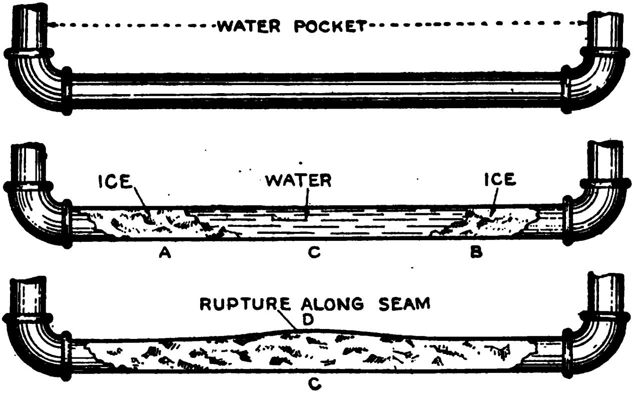
*NOTE.—According to the U. S. Bureau of Standards, 1915, the latent heat of fusion of ice is 143.57 B. t. u., however for ordinary calculations the value 144 is conveniently used.

4. Vaporization, or formation of steam at 212°.

During the process two changes of state have occurred:

1. *Solid to liquid* (ice to water) at freezing point 32° Fahr.;
2. Liquid to gas (water to steam) at boiling point 212° Fahr.

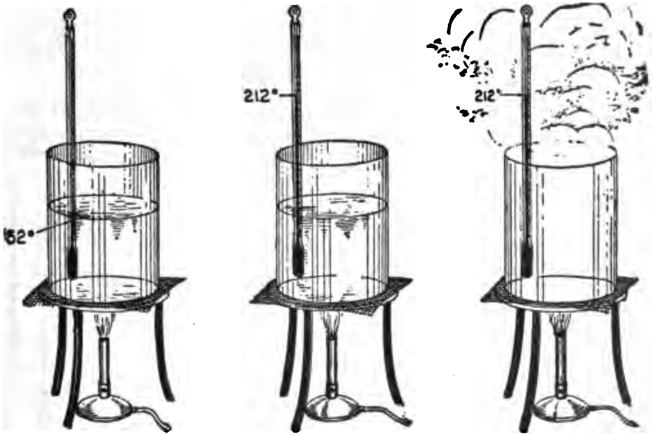
To effect these two changes of state, a considerable amount of work is done, especially in the case of the second change from liquid to gas, the amounts required being



FIGS. 42 to 44.—The bursting of pipes during freezing weather, illustrating the effect of pressure upon the freezing point. In draining pipes exposed to prevent freezing, care should be taken to remove all the water out of any water pockets that may exist, such as shown in fig. 42. The bursting of a pipe due to water in a pocket is illustrated in figs. 43 and 44, which show fig. 42 in section. Assuming the pocket to be full of water in freezing weather, it sometimes happens that the water at A and B, will freeze before it does at C, thus forming two slugs of ice enclosing the water C. When C, freezes, there being no room for expansion, the pipe bursts as indicated at D. *The popular impression* that pipes will burst at or very little below 32° Fahr., is erroneous. In fact the enormous pressure required to burst so called wrought iron pipe is not generally known, nor the effect of the pressure on the freezing point. For instance, the average bursting pressure of one-half inch standard pipe is 14,000 lbs., or 911.5 atmospheres per sq. in. and since the freezing point is lowered .0133° Fahr. for each additional atmosphere, the freezing point required to burst one-half inch pipe is 32 — (911.5 × .0133) = 20° Fahr.; that is to say, it would require a temperature of 20° to burst a one-half inch pipe of average strength by freezing.

1. To melt the ice . . . . 143.57 B. t. u.
2. To change the water at 212° Fahr. into steam of the same temperature . . . . 970.4 B. t. u.

It will be noted from these two items that it takes over five times as much heat to evaporate water at 212° Fahr. into steam of the same temperature, as it does to heat the water from the freezing point to the boiling point. That is to raise the temperature of the water from the freezing point to the boiling point requires 212 — 32 = 180 B. t. u., and from item 2, which represents the *latent heat of steam*, 970.4 B. t. u. are required to evaporate the water at 212° into steam of the same temperature. From which



FIGS. 45 to 47.—Experiment illustrating the ***latent heat of steam*** or *the considerable amount of heat which must be added to water at the boiling point to convert it into steam at the same temperature.* In fig. 45, suppose the glass vessel to contain one pound of water at 32° Fahr., and heat be transferred to it, as indicated by the bunsen burner, at such rate that its temperature is raised to the boiling point 212° in five minutes. In this time the water has received 212 — 32 = 180 heat units. Now, if the heat supply be continued at the same rate, it will require (since the latent heat of steam at atmospheric pressure is 970.4 heat units) 970.4 ÷ 180 = 5.39 times as long, or 5.39 × 5 = 26.95 minutes to convert the pound of water at 212° (fig. 46) into steam at the same temperature as indicated by the empty beaker in fig. 47. ***That is to say, it takes over five times as much heat to convert water at 212° into steam at the same temperature as it does to raise the same amount of water from the freezing point*** 32° to 212°. This experiment can be easily performed with water at ordinary temperature say 60°. In this case, if it take five minutes to raise its temperature to 212°, it would require 970.4 ÷ (212 — 60) = 6.38 times as long or 6.38 × 5 = 31.9 minutes to evaporate the water at 212°. It is thus seen that the latent heat is the big item in steam making. In the well known "***naphtha launch***" and "*alco-vapor launch*," naphtha and alcohol were used respectively in the boilers in the place of water because of the excessive latent heat of the latter. This with a given heating surface or weight (an important factor in marine construction) more power could be developed with the above mentioned liquids than with water, because of their relative low latent heat of evaporation. For instance, for alcohol the latent heat is 364.3 heat units or only a little over one-third of that of water. Operators of *alco-vapor launches* who have tried using water in the boiler can appreciate the considerable difference in the results obtained. From experiments made by the Gas Engine and Power Co., builders of the ***naphtha launches***, it was claimed that the power obtained on the brake was in the ratio of about 5 to 9 for steam and naphtha, that is, the same quantity of heat was turned into nearly twice as much work by the expansion of naphtha vapor as by the expansion of steam under the same conditions. Some of the results obtained during the tests were: ***1,*** with ***steam,*** mean pressure 37.99 lbs., r.p.m., 312.6; ***2,*** with ***naphtha,*** mean pressure 55.8, r.p.m., 552.2.

it requires 970.4÷180=5.39 times as much heat to evaporate the water as it does to heat the water between the limits given. This relation can be approximately determined by the interesting experiment shown in figs. 45 to 47. The heat which must be supplied during the process of evaporation has been expended in two ways.

1. In overcoming the molecular resistance of the medium H_2O in changing its state from a liquid to a gas;

2. In making room for itself against the pressure of the atmosphere, that is, doing *external work*. These two amounts of heat are called respectively

1. The ***internal latent heat;***
2. The ***external latent heat,***

both making up the ***latent heat of steam.***

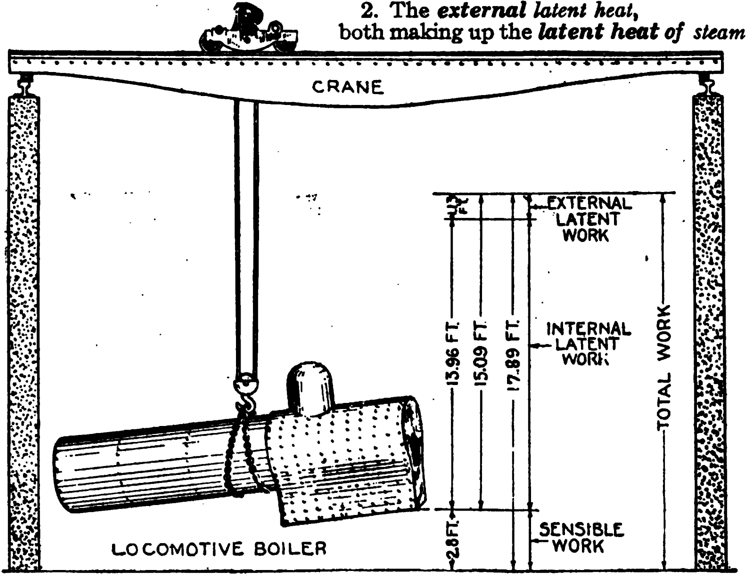


FIG. 48.—The work done in the formation of steam from water at 32° Fahr. In converting one pound of water at 32° into steam at 212°, 180 heat units are required to raise the temperature of the water to 212°; 897.51 heat units are absorbed by the water at 212° before a change of state takes place, and 72.89 heat units are required for the work to be done on the atmosphere to make room for the steam. These three items are known respectively as: 1, the **sensible heat**; 2, the **internal latent heat**, and 3, the **external latent heat.** The mechanical equivalent of one heat unit being 777.52 ft. lbs., the respective amounts of work corresponding to the ***sensible, internal,*** and ***external*** latent heats are 139,954 ft. lbs., 697,832 ft. lbs., and 56,674 ft. lbs. To grasp the significance of this, consider a locomotive boiler weighing, say 50,000 lbs. being lifted by a crane. The work done in lifting the boiler is equal to its weight in pounds multiplied by the distance raised in feet. Accordingly, for ***item 1,*** the work done is equivalent to raising the boiler 139,954÷50,000=2.8 ft.; for ***item*** 2, equivalent to 697,832÷50,000=13.96 ft., and for ***item 3***, equivalent to 56,674÷50,000=1.13 ft. Also the total work done in changing a pound of water at 32° into steam at 212°, or the sum of the three items is equivalent to raising the boiler 2.8+13.96+1.03=17.89 ft.

The author does not agree with the generally accepted calculation for the external latent heat, or external work of vaporization and holds that it is wrong in principle. The common method of calculating this work is based on the assumption that the amount of atmosphere displaced per pound of steam, is equal to the volume of one pound of saturated steam at the pressure under which it is formed; it is just this point wherein the error lies, as will now be shown. The volume of one pound of water at 212° atmospheric pressure, is 28.88 cu. ins. Now, if this water be placed in a long cylinder, having a cross sectional area of 144 sq. ins. it will occupy a depth of .0167 ft.

If a piston (assumed to have no weight and to move without friction) be placed on top of the water as in stage *4* (fig. 36), and heat applied, vaporization will begin, and when all the water has been changed into saturated steam, the volume has increased to 26.79 cu. ft., as in stage *6*, (fig. 38), that is, the volume of one pound of saturated steam at atmospheric pressure is 26.79 cu. ft.

Since the area of the piston is 1 sq. ft., the linear distance from the bottom of the cylinder to the piston is 26.79 ft., but *the piston has not moved this distance.* The initial position of the piston being .0167 ft. above the bottom of the cylinder, its actual movement is

$$26.79 - .0167 = 26.7733 \text{ ft.}$$

Accordingly, the work done by the steam in moving the piston against the pressure of the atmosphere to make room for itself or

external work = *area piston* × *pressure of atmosphere* × *movement of piston*
= 144 sq. ins. × 14.7 lbs. per sq. ins. × 26.7733 ft.
= 56,673.72 ft. lbs.

The erroneous method of making this calculation is to consider the movement of the piston equal to the distance between the bottom of the cylinder and the piston, or 26.79 ft., which would give for the external work

$$144 \times 14.7 \times 26.79 = 56{,}709.07 \text{ ft. lbs.}$$

being in excess of the true amount by

$$56{,}709.07 - 56{,}673.72 = 35.35 \text{ ft. lbs.}$$

or

$$.0167 \text{ ft.} \times 144 \text{ sq. ins.} \times 14.7 = 35.35 \text{ ft. lbs.}$$

Motion is purely a relative matter, and accordingly something must be regarded as being stationary as a basis for defining motion; hence the question:

Is the movement of the piston in stage 6 (fig. 38) *to be referred to a* ***stationary*** *water level or to a* ***receding*** *water level?*

The author holds that the movement of the piston referred to a stationary water level gives the true displacement of the air and is accordingly the proper basis for calculating the external work. It must be evident that since the water already existed at the beginning of vaporization, the atmosphere was already displaced to the extent of the volume occupied

From Ice to Steam

	British thermal units		Mechanical equivalent in foot pounds		Volume
1. *Latent Heat of Fusion*					*In cu. ins.*
a To *melt* one lb of ice at 32° Fahr *without* changing its temperature	143.57§				
b ***Mechanical equivalent*** = 143.57 × 777.52 =			111,675.62		
c ***Volume*** { of one lb of ice at 32° Fahr.					30.067
" " " water " " "					27.7
Values given below are "above 32° Fahr."‡					
2. *Sensible Heat*					
a. To raise the temperature of one lb of water from 32° to 212° Fahr. (212°—32°) =		180.*			
b ***Mechanical equivalent*** = 180 × 777.52 = .				139,953.6	
c ***Volume*** { of one lb of water at 39.1° Fahr., point of maximum density					27.68
" " " " " 212° " boiling point (atmospheric pressure)					28.88
3. *Latent Heat of Evaporation*					
Being the heat necessary to convert one lb. of water at 212° Fahr into steam of the same temperature and consisting of:					
a. The internal latent heat					
1 To overcome the molecular attraction, that is, to separate the particles of the water in the formation of steam =	897.51				
2. ***Mechanical equivalent*** = 897.51 × 777.52 =			697,831 96		
b. The external latent heat					*In cu. ft.*
1. To overcome atmospheric pressure, that is, to push back the atmosphere in order to make room for the steam = mechanical equivalent (item 3 b 2) ÷ 777.52 =	72.89				
2 ***Mechanical equivalent*** = 144 sq. ins. × 14.7 lbs. × (26.79 ft.—.0167 ft)† =			56,673.72		
3 ***Volume*** { of one lb. of water at 212° Fahr					0167
" " " " steam " " "					26.79
c. Total latent heat					
1 For evaporation at 212° Fahr., the total latent heat, called simply ***latent heat*** = item 3 a 1 + item 3 b 1 =		970.4*			
2 ***Mechanical equivalent*** = item 3 a 2 + item 3 b 2 =				754,505.7	
4. *Total Heat*					
a. Above 32° Fahr in the ***water*** = item 2 a	180				
b. " 32° " " " ***steam*** = item 2 a + item 3 c 1 =		1,150.4*			
c. ***Mechanical equivalent*** { for the ***water*** = item 2 b =			139,953.6		
" " ***steam*** = item 2 b + item 3 c 2 =				894,459.3	

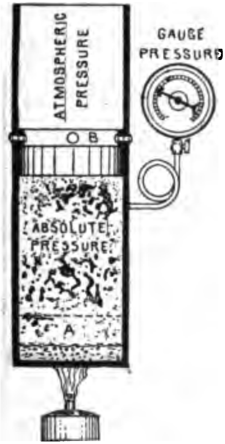
§U. S. Bureau of Standards (1915). *According to Marks and Davis. †As calculated by the author (see page 31). ‡Some early steam tables give values "*above 0° Fahr.*"

by the water, and therefore this displacement must not be considered as contributing to the external work done by the steam during its formation. The amount of error (35.35 ft. lbs.) of the common calculation, though very small, is an appreciable amount; its equivalent in heat units is

$$35.35 \div 777.52 = .0455 \text{ B. t. u.}$$

The thermal equivalent of the external work is

$$56{,}673.72 \div 777.52 = 72.89 \text{ B. t. u.}$$



Condensation of Steam.—*When the temperature of steam becomes less than that corresponding to its pressure,* **condensation** *takes place,* that is, it ceases to exist as steam and becomes water.

FIG. 49.—The generation of steam at pressures above the atmosphere. If a ring B, be riveted in a cylinder to limit the movement of a piston resting, at the beginning of the experiment, on top of a small quantity of water (as indicated by dotted lines A, and heat be applied, the piston (assumed to have no weight) will rise as steam is formed at atmospheric pressure until it comes in contact with the ring B. Additional heat will cause the pressure of the steam to increase in a definite rate corresponding to the temperature until all the water is evaporated, the cylinder being now filled with *saturated steam.* The pressure of this saturated steam will depend on the relation between its volume and the volume of the water from which it was generated. If more heat be now added, the temperature of the steam will increase above that due to its pressure, and the steam becomes *superheated.* Removing the heat supply, the temperature of the gas will gradually diminish, and it loses its superheat and returns to the saturated condition, at which point condensation begins, the pressure and temperature during these changes gradually falling. Condensation continuing until all the steam has condensed, the piston returning to its initial position A. If, during the cooling process, the piston be fastened at the ring B, the pressure of the steam will become less than the atmospheric pressure outside when the temperature falls below 212° Fahr., forming a so called vacuum. The degree of vacuum now increases, or in other words, the pressure under the cylinder or *absolute pressure* becomes less and less until, when all the steam is condensed, it becomes approximately zero, or 14.72 lbs. lower than the pressure of the atmosphere or *gauge pressure,* (assuming the barometer reads 30 inches). The pressure remains a little above zero because of the small percentage of air originally contained in the water, which does not recombine with it when the steam condenses, that is, a *perfect vacuum* is not formed *because of this air,* necessitating, in the case of condensing engines, an air or so called vacuum pump.

Thus in fig. 50, if cold water be poured on the inverted flask, containing steam and water, the steam will be cooled *below* its temperature corresponding to its pressure (as given in the steam table) and some of it will condense. This will cause a reduction of pressure because the volume of steam is greatly diminished after condensation.* On account of the reduction in pressure, the water will again boil vigorously until enough steam has been formed to increase the pressure to correspond with the boiling point.

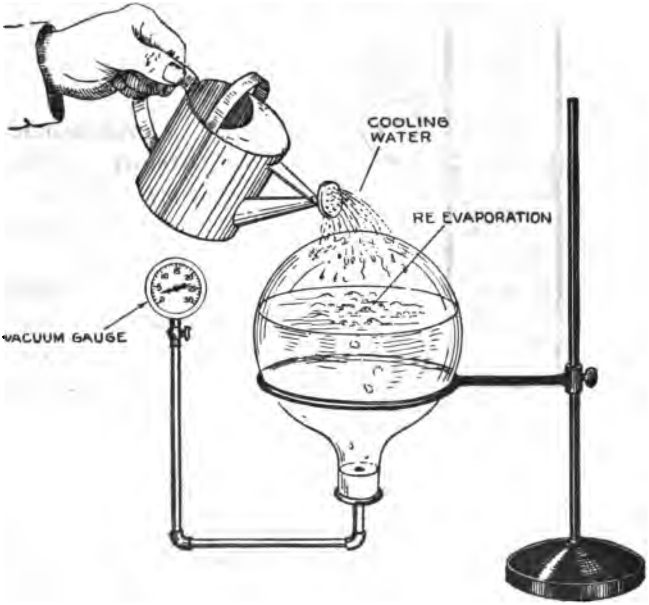


FIG. 50.—Lowering the boiling point by diminishing the pressure. Fill a round bottomed flask with water and boil. After it has boiled some time, until the air has been drawn out of the flask by the steam, insert a rubber stopper, having fitted to it a connection leading to a vacuum gauge and invert the flask as shown. The vacuum gauge will now read zero. Now, if some cold water be poured over the flask, the temperature will fall rapidly and some of the steam will condense, thus lowering the pressure within the flask, that is, the vacuum gauge will read 5 or 10 inches indicating a vacuum. The reduced pressure disturbs the equilibrium between pressure and temperature and the water will boil until equilibrium is again restored. The operation may be repeated several times without reheating, the pressure gradually falling each time. At the city of Quito, Ecuador, water boils at 194° Fahr., and on the top of Mt. Blanc at 183°. Again, in a steam boiler in which the pressure is 200 lbs., the boiling point is 387.7°.

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*NOTE.—It should be remembered that 1 cu. ft. of steam at atmospheric pressure is reduced in volume after condensation to approximately 1 cu. in.

A second application of cold water will again cause the water to boil, the result being the same so long as the water in the flask is at a higher temperature than the water applied outside.

The greater the difference in temperature, the more vigorous will the water boil. This illustrates an important effect in the behavior of steam in a steam engine, namely, *re-evaporation* which will be later explained.

Ques. If a closed flask containing steam and water be allowed to stand for a length of time, what happens?

Ans. The atmosphere being at a lower temperature than that inside the flask, will abstract heat from the steam and water, *but the heat will leave the steam quicker than the water*. The result is a continuous condensation of the steam and re-evaporation of the water, during which process the temperature of the whole mass and the boiling point is gradually lowered until the temperature inside of the flask is the same as that outside. This process is accompanied by a gradual decrease in pressure.

Ques. Why does the pressure fall?

Ans. Because the temperature falls.

There is a fixed pressure for each degree of temperature of the water as tabulated in the steam tables.

Ques. Can this pressure be reduced to zero by reducing the temperature of the water?

Ans. It could, if the mass could be cooled to 459.4° below zero Fahr.*, but at ordinary temperatures the pressure could not be reduced to zero. Water contains a small amount of air which it gives up when evaporated; this of itself would prevent the pressure falling to zero, if all the steam were condensed. If the contents of the flask be cooled to 80° there would be inside the flask a pressure of one-half pound per square inch,

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*NOTE.—459.6° below zero Fahr., as previously explained, is a point called the *absolute zero*. A perfect gas contracts in volume a definite amount for each degree in temperature it is cooled. The absolute zero then is the point to which a perfect gas must be cooled to reduce its volume to nothing.

if the water be cooled to 32°, the pressure would be .089 pounds. There is then always some pressure inside the flask, the intensity of which depends upon the temperature of the water.

Ques. How was this principle first made use of?

Ans. The early engineers discovered that by condensation the pressure of the atmosphere is made available for doing work.

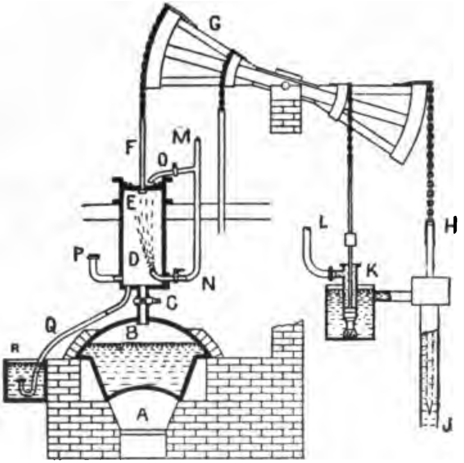

FIG. 51.—Newcomen's atmospheric engine. ***The parts are:*** A, furnace; B, boiler; C, valve; D, cylinder; E, piston; F, piston rod; G, walking beam; H, pump rod; J, pump cylinder; K, pump barrel; L, injection water pump; M, injection water pipe; N, injection valve; O, water supply cock to seal piston; P, air check or snifting valve; Q, injection water discharge pipe; R, hot well. In Newcomen's engine, the piston was attached by rod and chain to one end of the walking-beam, and the pump rod to the other end. The pump rod was heavy enough to sink it in the barrel and raise the steam piston, or else a weight was added. The periphery of the piston was covered with leather and kept air tight by water above it, admitted through cock O. The cylinder D, was placed above the boiler B, and steam was admitted to it through the cock C, which was tended by hand, the strokes being slow. ***At starting,*** the air from the cylinder, displaced by the steam, passed out through the pipe which proceeds from the bottom of the cylinder, and issued at the valve P, which opened upwardly. This is the blow valve or ***snifting valve*** of the engine. The cock C, being then closed, shuts off the steam, and the cock N, being opened, allows injection water to enter the cylinder from injection pump K, through pipe M. The water, being condensed into about $^1/_{1728}$ of its bulk, formed a nearly perfect vacuum, and the atmospheric pressure of 14.7 pounds to the square inch bearing upon the piston depressed the latter, and consequently raised the pump rod, the weight (if there be any), and the load of water. The downward stroke only of the piston was used effectively. The water of injection and condensation passed by the pipe Q, leading from the bottom of the cylinder to the hot well R, issuing through a check valve, and was used to feed the boiler. It will be observed that the piston and pump rod are merely suspended by chains; the action of each is to pull, not push, and a stiff connection was not necessary. At first Newcomen adopted Savery's plan of external condensation, but a faulty cylinder having admitted water internally, the condensation was more rapid with increased effect from the engine.

NOTE.—The taps which answered as valves in the Newcomen engine required the most unremitting attention of the person in charge, to introduce steam into the cylinder to lift the piston, or the shower of cold water which was to condense the steam and cause the depression of the piston by the atmospheric pressure above it. A Cornish boy, named Potter, in order to have some time for play conceived and put in execution the idea of connecting the beam to the handle of the taps, so as to work them automatically. Hence the valve motion. For the first time, the engine worked by itself. With the exception of Smeaton's improvements in details, the Newcomen engine remained in the state to which its inventor had brought it, from 1710 to 1764, about which time Watt appeared.

Ques. What is the chief objection to this engine?

Ans. It was discovered by James Watt in 1763 that there was a large waste of steam in the cylinder owing to condensation of the steam through contact with the cold wet cylinder walls.

Ques. How did Watt overcome this defect?

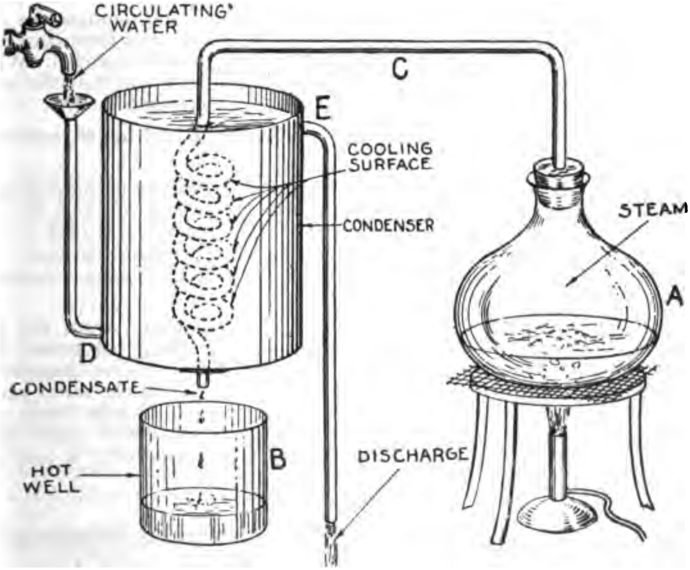

FIG. 52.—The condensation of steam. If water be boiled in a flask A, and the steam thus produced led off through pipe C, having a coiled section surrounded by cold water, it will here be cooled below the boiling point and will therefore condense, the *condensate* passing out into the receptacle B, as water. The cooling or "circulating" water enters the condenser at the lowest point D, and leaving at the highest point E.

Ans. He invented a separate chamber in which the condensation took place. The steam was passed from the cylinder into this chamber, called the *condenser* where it was condensed by contact with cold water without the need of cooling the cylinder itself.

The elementary condenser shown in fig. 52 illustrates the method of condensing steam after it leaves the engine cylinder by *bringing it in contact with a cold metallic surface.* Such method is called *surface condensation,* and the apparatus, a *surface condenser.*

In the figure, the flask A, placed over a Bunsen burner, is fitted with a rubber cork and tube C, part of which is coiled and surrounded by flowing cold water. A glass vessel or beaker B, is placed under the end of the coil as shown.

When water is boiled in A, the steam thus formed will pass off through C, and in traversing the coiled part it is cooled below the temperature corresponding to its pressure and *condenses.* The water thus formed or *condensate* passes in drops from the end of the tube C, into the vessel B. When all the water has been evaporated from A, it will be found that the same weight of water originally placed in A, has been deposited in B.

The condensation of steam may also be illustrated by fig. 49, assuming the supply of heat to be removed.

If the piston be pushed downward, the tendency will be to increase both the pressure and temperature of the steam. However, if there be any increase in the temperature of the steam it is immediately cooled by the water which causes condensation and keeps the pressure constant. In fact all the steam could be condensed by pushing the piston downward until it rested on the surface of the water.

If the piston be now returned to its original position, steam will immediately form and fill the space, the pressure remaining constant. The reason for this is that unless the empty space formed by the receding piston be immediately filled, the pressure will fall, thus exposing the water which is at 212° to a pressure lower than the boiling point. Under these conditions the water boils until the empty space is filled with steam of a density corresponding to the boiling point, thus preserving a constant pressure.

How to Use the Steam Table.—The various properties of *saturated steam* are usually presented in tabulated form for convenience in making calculations. The values of the properties of steam here given are condensed from Marks and Davis steam tables which are now (1917) generally accepted as the standard, and are the most accurate that have yet been published.

In the first column is given the gauge pressure, and in the second, the absolute pressure. The second column, then, is made up by adding 14.7 lbs. to the pressures given in the first column. Before using a steam table, the difference between gauge and absolute pressure should be thoroughly understood.

The third column gives the temperature in degrees Fahrenheit, beginning with the freezing point 32°, which for convenience is taken as *the temperature of no heat*.

Column four gives the total heat above 32° in each pound of water at the different pressures; similarly in column five is given the total heat above 32° for each pound weight of steam.

The latent heat in the next column is clearly the difference between the heat in the steam and the heat in the water, or column 5—column 4.

The relative volume of the steam is given in column seven; for instance, one pound of water at 212° will occupy 26.79 cu. ft. when evaporated into steam at the same temperature.

Column eight gives the weight per cu. ft., and the last two columns the entropy values.

The following examples illustrate how to use the steam table:

Example.—How many heat units are saved in heating 25 lbs. of feed water from 90 to 202°?

In column 4, total heat in the water at 201.96° = 169.9
In column 4, total heat in the water at 90° = 58.0
Heat units saved per lb. of feed water = 111.9
Total heat units saved = 111.9×25 = 2,797.5

Example.—What is the weight of 20 cu. ft. of steam at 150 lbs. absolute pressure?

The weight of 1 cu. ft. steam at 150 lbs. abs. is given in column 9 at .332 lb. Twenty cu. ft. then will weigh: .332×20 = 6.64 lbs.

Example.—How much more heat is required to generate 26 lbs. of steam at 150 lbs. abs., than at 90 lbs. abs.

In column 5 total heat in steam at 150 lbs. abs. = 1,193.4
In column 5 total heat in steam at 90 lbs. abs. = 1,184.4
Excess heat required per pound (weight) = 9 B. t. u.
Total for 26 lbs. = 9×26 = 23.4 B. t. u.

Example.—How much heat is absorbed by the cooling water, if a condensing engine exhaust 17 lbs. of steam per hour at a terminal pressure of 18 lbs. absolute into a 28 inch vacuum.

In column 5, total heat in the steam at 18 lbs. abs. = 1,154.20
In column 4, total heat in the water with 28″ vacuum = 67.97
Heat to be absorbed per lb. of steam . . . . = 1,086.23
Total heat absorbed by the cooling water per hour

1,086.23×17 = 18,465.9 B. t. u.

Properties of Saturated Steam

Condensed from Marks and Davis' Steam Tables and Diagrams, 1909, by permission of the publishers, Longmans, Green & Co.

Vacuum, Inches of Mercury.	Absolute Pressure, Lbs. per Sq. In.	Temperature, Fahrenheit.	Total Heat above 32° F		Latent Heat, L = H—h Heat-Units.	Volume, Cu. Ft. in 1 Lb. of Steam.	Weight of 1 Cu. Ft. Steam, Lb.	Entropy of the Water.	Entropy of Evaporation.
			In the Water h Heat-Units.	In the Steam H Heat-Units.					
29.74	0.0886	32	0.00	1073.4	1073.4	3294	0.000304	0.0000	2.1832
29.67	0.1217	40	8.05	1076.9	1068.9	2438	0.000410	0.0162	2.1394
29.56	0.1780	50	18.08	1081.4	1063.3	1702	0.000587	0.0361	2.0865
29.40	0.2562	60	28.08	1085.9	1057.8	1208	0.000828	0.0555	2.0358
29.18	0.3626	70	38.06	1090.3	1052.3	871	0.001148	0.0745	1.9868
28.89	0.505	80	48.03	1094.8	1046.7	636.8	0.001570	0.0932	1.9398
28.50	0.696	90	58.00	1099.2	1041.2	469.3	0.002131	0.1114	1.8944
28.00	0.946	100	67.97	1103.6	1035.6	350.8	0.002851	0.1295	1.8505
27.88	1	101.83	69.8	1104.4	1034.6	333.0	0.00300	0.1327	1.8427
25.85	2	126.15	94.0	1115.0	1021.0	173.5	0.00576	0.1749	1.7431
23.81	3	141.52	109.4	1121.6	1012.3	118.5	0.00845	0.2008	1.6840
21.78	4	153.01	120.9	1126.5	1005.7	90.5	0.01107	0.2198	1.6416
19.74	5	162.28	130.1	1130.5	1000.3	73.33	0.01364	0.2348	1.6084
17.70	6	170.06	137.9	1133.7	995.8	61.89	0.01616	0.2471	1.5814
15.67	7	176.85	144.7	1136.5	991.8	53.56	0.01867	0.2579	1.5582
13.63	8	182.86	150.8	1139.0	988.2	47.27	0.02115	0.2673	1.5380
11.60	9	188.27	156.2	1141.1	985.0	42.36	0.02361	0.2756	1.5202
9.56	10	193.22	161.1	1143.1	982.0	38.38	0.02606	0.2832	1.5042
7.52	11	197.75	165.7	1144.9	979.2	35.10	0.02849	0.2902	1.4895
5.49	12	201.96	169.9	1146.5	976.6	32.36	0.03090	0.2967	1.4760
3.45	13	205.87	173.8	1148.0	974.2	30.03	0.03330	0.3025	1.4639
1.42	14	209.55	177.5	1149.4	971.9	28.02	0.03569	0.3081	1.4523
lbs. gauge	14.70	212	180.0	1150.4	970.4	26.79	0.03732	0.3118	1.4447
0.3	15	213.0	181.0	1150.7	969.7	26.27	0.03806	0.3133	1.4416
1.3	16	216.3	184.4	1152.0	967.6	24.79	0.04042	0.3183	1.4311
2.3	17	219.4	187.5	1153.1	965.6	23.38	0.04277	0.3229	1.4215
3.3	18	222.4	190.5	1154.2	963.7	22.16	0.04512	0.3273	1.4127
4.3	19	225.2	193.4	1155.2	961.8	21.07	0.04746	0.3315	1.4045
5.3	20	228.0	196.1	1156.2	960.0	20.08	0.04980	0.3355	1.3965
6.3	21	230.6	198.8	1157.1	958.3	19.18	0.05213	0.3393	1.3887
7.3	22	233.1	201.3	1158.0	956.7	18.37	0.05445	0.3430	1.3811
8.3	23	235.5	203.8	1158.8	955.1	17.62	0.05676	0.3465	1.3739
9.3	24	237.8	206.1	1159.6	953.5	16.93	0.05907	0.3499	1.3670
10.3	25	240.1	208.4	1160.4	952.0	16.30	0.0614	0.3532	1.3604
11.3	26	242.2	210.6	1161.2	950.6	15.72	0.0636	0.3564	1.3542
12.3	27	244.4	212.7	1161.9	949.2	15.18	0.0659	0.3594	1.3483

Properties of Saturated Steam—*Continued*

Gauge Pressure, Lbs. per Sq. In.	Absolute Pressure, Lbs. per sq. in.	Temperature, Fahrenheit.	Total Heat above 32° F		Latent Heat, L= H—h Heat-Units.	Volume, Cu. Ft. in 1 Lb. of Steam.	Weight of 1 Cu. Ft. Steam, Lb.	Entropy of the Water.	Entropy of Evaporation.
			In the water h Heat-Units.	In the Steam H Heat-Units.					
13.3	28	246.4	214.8	1162.6	947.8	14.67	0.0682	0.3623	1.3425
14.3	29	248.4	216.8	1163.2	946.4	14.19	0.0705	0.3652	1.3367
15.3	30	250.3	218.8	1163.9	945.1	13.74	0.0728	0.3680	1.3311
16.3	31	252.2	220.7	1164.5	943.8	13.32	0.0751	0.3707	1.3257
17.3	32	254.1	222.6	1165.1	942.5	12.93	0.0773	0.3733	1.3205
18.3	33	255.8	224.4	1165.7	941.3	12.57	0.0795	0.3759	1.3155
19.3	34	257.6	226.2	1166.3	940.1	12.22	0.0818	0.3784	1.3107
20.3	35	259.3	227.9	1166.8	938.9	11.89	0.0841	0.3806	1.3060
21.3	36	261.0	229.6	1167.3	937.7	11.58	0.0863	0.3832	1.3014
22.3	37	262.6	231.3	1167.8	936.6	11.29	0.0886	0.3855	1.2969
23.3	38	264.2	232.9	1168.4	935.5	11.01	0.0908	0.3877	1.2925
24.3	39	265.8	234.5	1168.9	934.4	10.74	0.0931	0.3899	1.2882
25.3	40	267.3	236.1	1169.4	933.3	10.49	0.0953	0.3920	1.2841
26.3	41	268.7	237.6	1169.8	932.2	10.25	0.0976	0.3941	1.2800
27.3	42	270.2	239.1	1170.3	931.2	10.02	0.0998	0.3962	1.2759
28.3	43	271.7	240.5	1170.7	930.2	9.80	0.1020	0.3982	1.2720
29.3	44	273.1	242.0	1171.2	929.2	9.59	0.1043	0.4002	1.2681
30.3	45	274.5	243.4	1171.6	928.2	9.39	0.1065	0.4021	1.2644
31.3	46	275.8	244.8	1172.0	927.2	9.20	0.1087	0.4040	1.2607
32.3	47	277.2	246.1	1172.4	926.3	9.02	0.1109	0.4059	1.2571
33.3	48	278.5	247.5	1172.8	925.3	8.84	0.1131	0.4077	1.2536
34.3	49	279.8	248.8	1173.2	924.4	8.67	0.1153	0.4095	1.2502
35.3	50	281.0	250.1	1173.6	923.5	8.51	0.1175	0.4113	1.2468
36.3	51	282.3	251.4	1174.0	922.6	8.35	0.1197	0.4130	1.2435
37.3	52	283.5	252.6	1174.3	921.7	8.20	0.1219	0.4147	1.2402
38.3	53	284.7	253.9	1174.7	920.8	8.05	0.1241	0.4164	1.2370
39.3	54	285.9	255.1	1175.0	919.9	7.91	0.1263	0.4180	1.2339
40.3	55	287.1	256.3	1175.4	919.0	7.78	0.1285	0.4196	1.2309
41.3	56	288.2	257.5	1175.7	918.2	7.65	0.1307	0.4212	1.2278
42.3	57	289.4	258.7	1176.0	917.4	7.52	0.1329	0.4227	1.2248
43.3	58	290.5	259.8	1176.4	916.5	7.40	0.1350	0.4242	1.2218
44.3	59	291.6	261.0	1176.7	915.7	7.28	0.1372	0.4257	1.2189
45.3	60	292.7	262.1	1177.0	914.9	7.17	0.1394	0.4272	1.2160
46.3	61	293.8	263.2	1177.3	914.1	7.06	0.1416	0.4287	1.2132
47.3	62	294.9	264.3	1177.6	913.3	6.95	0.1438	0.4302	1.2104
48.3	63	295.9	265.4	1177.9	912.5	6.85	0.1460	0.4316	1.2077
49.3	64	297.0	266.4	1178.2	911.8	6.75	0.1482	0.4330	1.2050
50.3	65	298.0	267.5	1178.5	911.0	6.65	0.1503	0.4344	1.2024
51.3	66	299.0	268.5	1178.8	910.2	6.56	0.1525	0.4358	1.1998
52.3	67	300.0	269.6	1179.0	909.5	6.47	0.1547	0.4371	1.1972

Properties of Saturated Steam—*Continued*

Gauge Pressure, Lbs. per Sq. In.	Absolute Pressure, Lbs. per sq. in.	Temperature, Fahrenheit.	Total Heat above 32° F		Latent Heat, L= H—h Heat-Units.	Volume, Cu. Ft. in 1 Lb. of Steam.	Weight of 1 Cu. Ft. Steam, Lb.	Entropy of the Water.	Entropy of Evaporation.
			In the water h Heat-Units.	In the Steam H Heat-Units.					
53.3	68	301.0	270.6	1179.3	908.7	6.38	0.1569	0.4385	1.1946
54.3	69	302.0	271.6	1179.6	908.0	6.29	0.1590	0.4398	1.1921
55.3	70	302.9	272.6	1179.8	907.2	6.20	0.1612	0.4411	1.1896
56.3	71	303.9	273.6	1180.1	906.5	6.12	0.1634	0.4424	1.1872
57.3	72	304.8	274.5	1180.4	905.8	6.04	0.1656	0.4437	1.1848
58.3	73	305.8	275.5	1180.6	905.1	5.96	0.1678	0.4449	1.1825
59.3	74	306.7	276.5	1180.9	904.4	5.89	0.1699	0.4462	1.1801
60.3	75	307.6	277.4	1181.1	903.7	5.81	0.1721	0.4474	1.1778
61.3	76	308.5	278.3	1181.4	903.0	5.74	0.1743	0.4487	1.1755
62.3	77	309.4	279.3	1181.6	902.3	5.67	0.1764	0.4499	1.1732
63.3	78	310.3	280.2	1181.8	901.7	5.60	0.1786	0.4511	1.1710
64.3	79	311.2	281.1	1182.1	901.0	5.54	0.1808	0.4523	1.1687
65.3	80	312.0	282.0	1182.3	900.3	5.47	0.1829	0.4535	1.1665
66.3	81	312.9	282.9	1182.5	899.7	5.41	0.1851	0.4546	1.1644
67.3	82	313.8	283.8	1182.8	899.0	5.34	0.1873	0.4557	1.1623
68.3	83	314.6	284.6	1183.0	898.4	5.28	0.1894	0.4568	1.1602
69.3	84	315.4	285.5	1183.2	897.7	5.22	0.1915	0.4579	1.1581
70.3	85	316.3	286.3	1183.4	897.1	5.16	0.1937	0.4590	1.1561
71.3	86	317.1	287.2	1183.6	896.4	5.10	0.1959	0.4601	1.1540
72.3	87	317.9	288.0	1183.8	895.8	5.05	0.1980	0.4612	1.1520
73.3	88	318.7	288.9	1184.0	895.2	5.00	0.2001	0.4623	1.1500
74.3	89	319.5	289.7	1184.2	894.6	4.94	0.2023	0.4633	1.1481
75.3	90	320.3	290.5	1184.4	893.9	4.89	0.2044	0.4644	1.1461
76.3	91	321.1	291.3	1184.6	893.3	4.84	0.2065	0.4654	1.1442
77.3	92	321.8	292.1	1184.8	892.7	4.79	0.2087	0.4664	1.1423
78.3	93	322.6	292.9	1185.0	892.1	4.74	0.2109	0.4674	1.1404
79.3	94	323.4	293.7	1185.2	891.5	4.69	0.2130	0.4684	1.1385
80.3	95	324.1	294.5	1185.4	890.9	4.65	0.2151	0.4694	1.1367
81.3	96	324.9	295.3	1185.6	890.3	4.60	0.2172	0.4704	1.1348
82.3	97	325.6	296.1	1185.8	889.7	4.56	0.2193	0.4714	1.1330
83.3	98	326.4	296.8	1186.0	889.2	4.51	0.2215	0.4724	1.1312
84.3	99	327.1	297.6	1186.2	888.6	4.47	0.2237	0.4733	1.1295
85.3	100	327.8	298.3	1186.3	888.0	4.429	0.2258	0.4743	1.1277
87.3	102	329.3	299.8	1186.7	886.9	4.347	0.2300	0.4762	1.1242
89.3	104	330.7	301.3	1187.0	885.8	4.268	0.2343	0.4780	1.1208
91.3	106	332.0	302.7	1187.4	884.7	4.192	0.2336	0.4798	1.1174
93.3	108	333.4	304.1	1187.7	883.6	4.118	0.2429	0.4816	1.1141
95.3	110	334.8	305.5	1188.0	882.5	4.047	0.2472	0.4834	1.1108
97.3	112	336.1	306.9	1188.4	881.4	3.978	0.2514	0.4852	1.1076
99.3	114	337.4	308.3	1188.7	880.4	3.912	0.2556	0.4869	1.1045

Properties of Saturated Steam—*Continued*

Gauge Pressure, Lbs. per Sq. In.	Absolute Pressure, Lbs. per Sq. In.	Temperature, Fahrenheit.	Total Heat above 32° F		Latent Heat, L= H—*h* Heat-Units.	Volume, Cu. Ft. in 1 Lb. of Steam.	Weight of 1 Cu. Ft. Steam, Lb.	Entropy of the Water.	Entropy of Evaporation.
			In the Water *h* Heat-Units.	In the Steam H Heat-Units.					
101.3	116	338.7	309.6	1189.0	879.3	3.848	0.2599	0.4886	1.1014
103.3	118	340.0	311.0	1189.3	878.3	3.786	0.2641	0.4903	1.0984
105.3	120	341.3	312.3	1189.6	877.2	3.726	0.2683	0.4919	1.0954
107.3	122	342.5	313.6	1189.8	876.2	3.668	0.2726	0.4935	1.0924
109.3	124	343.8	314.9	1190.1	875.2	3.611	0.2769	0.4951	1.0895
111.3	126	345.0	316.2	1190.4	874.2	3.556	0.2812	0.4967	1.0865
113.3	128	346.2	317.4	1190.7	873.3	3.504	0.2854	0.4982	1.0837
115.3	130	347.4	318.6	1191.0	872.3	3.452	0.2897	0.4998	1.0809
117.3	132	348.5	319.9	1191.2	871.3	3.402	0.2939	0.5013	1.0782
119.3	134	349.7	321.1	1191.3	870.4	3.354	0.2981	0.5028	1.0755
121.3	136	350.8	322.3	1191.7	869.4	3.308	0.3023	0.5043	1.0728
123.3	138	352.0	323.4	1192.6	868.5	3.263	0.3065	0.5057	1.0702
125.3	140	353.1	324.6	1192.2	867.6	3.219	0.3107	0.5072	1.0675
127.3	142	354.2	325.8	1192.5	866.7	3.175	0.3150	0.5086	1.0649
129.3	144	355.3	326.9	1192.7	865.8	3.133	0.3192	0.5100	1.0624
131.3	146	356.3	328.0	1192.9	864.9	3.092	0.3234	0.5114	1.0599
133.3	148	357.4	329.1	1193.2	864.0	3.052	0.3276	0.5128	1.0574
135.3	150	358.5	330.2	1193.4	863.2	3.012	0.3320	0.5142	1.0550
137.3	152	359.5	331.4	1193.6	862.3	2.974	0.3362	0.5155	1.0525
139.3	154	360.5	332.4	1193.8	861.4	2.938	0.3404	0.5169	1.0501
141.3	156	361.6	333.5	1194.1	860.6	2.902	0.3446	0.5182	1.0477
143.3	158	362.6	334.6	1194.3	859.7	2.868	0.3488	0.5195	1.0454
145.3	160	363.6	335.6	1194.5	858.8	2.834	0.3529	0.5208	1.0431
147.3	162	364.6	336.7	1194.7	858.0	2.801	0.3570	0.5220	1.0409
149.3	164	365.6	337.7	1194.9	857.2	2.769	0.3612	0.5233	1.0387
151.3	166	366.5	338.7	1195.1	856.4	2.737	0.3654	0.5245	1.0365
153.3	168	367.5	339.7	1195.3	855.5	2.706	0.3696	0.5257	1.0343
155.3	170	368.5	340.7	1195.4	854.7	2.675	0.3738	0.5269	1.0321
157.3	172	369.4	341.7	1195.6	853.9	2.645	0.3780	0.5281	1.0300
159.3	174	370.4	342.7	1195.8	853.1	2.616	0.3822	0.5293	1.0278
161.3	176	371.3	343.7	1196.0	852.3	2.588	0.3864	0.5305	1.0257
163.3	178	372.2	344.7	1196.2	851.5	2.560	0.3906	0.5317	1.0235
165.3	180	373.1	345.6	1196.4	850.8	2.533	0.3948	0.5328	1.0215
167.3	182	374.0	346.6	1196.6	850.0	2.507	0.3989	0.5339	1.0195
169.3	184	374.9	347.6	1196.8	849.2	2.481	0.4031	0.5351	1.0174
171.3	186	375.8	348.5	1196.9	848.4	2.455	0.4073	0.5362	1.0154
173.3	188	376.7	349.4	1197.1	847.7	2.430	0.4115	0.5373	1.0134
175.3	190	377.6	350.4	1197.3	846.9	2.406	0.4157	0.5384	1.0114
177.3	192	378.5	351.3	1197.4	846.1	2.381	0.4199	0.5395	1.0095
179.3	194	379.3	352.2	1197.6	845.4	2.358	0.4241	0.5405	1.0076

Properties of Saturated Steam—*Continued*

Gauge Pressure, Lbs. per Sq. In.	Absolute Pressure, Lbs. per Sq. In.	Temperature, Fahrenheit.	Total Heat above 32° F		Latent Heat, L= H—h Heat-Units.	Volume, Cu. Ft. in 1 Lb. of Steam.	Weight of 1 Cu. Ft. Steam, Lb.	Entropy of the Water.	Entropy of Evaporation.
			In the Water h Heat-Units.	In the Steam H Heat-Units.					
181.3	196	380.2	353.1	1197.8	844.7	2.335	0.4283	0.5416	1.0056
183.3	198	381.0	354.0	1197.9	843.9	2.312	0.4325	0.5426	1.0038
185.3	200	381.9	354.9	1198.1	843.2	2.290	0.437	0.5437	1.0019
190.3	205	384.0	357.1	1198.5	841.4	2.237	0.447	0.5463	0.9973
195.3	210	386.0	359.2	1198.8	839.6	2.187	0.457	0.5488	0.9928
200.3	215	388.0	361.4	1199.2	837.9	2.138	0.468	0.5513	0.9885
205.3	220	389.9	363.4	1199.6	836.2	2.091	0.478	0.5538	0.9841
210.3	225	391.9	365.5	1199.9	834.4	2.046	0.489	0.5562	0.9799
215.3	230	393.8	367.5	1200.2	832.8	2.004	0.499	0.5586	0.9758
220.3	235	395.6	369.4	1200.6	831.1	1.964	0.509	0.5610	0.9717
225.3	240	397.4	371.4	1200.9	829.5	1.924	0.520	0.5633	0.9676
230.3	245	399.3	373.3	1201.2	827.9	1.887	0.530	0.5655	0.9638
235.3	250	401.1	375.2	1201.5	826.3	1.850	0.541	0.5676	0.9600
245.3	260	404.5	378.9	1202.1	823.1	1.782	0.561	0.5719	0.9525
255.3	270	407.9	382.5	1202.6	820.1	1.718	0.582	0.5760	0.9454
265.3	280	411.2	386.0	1203.1	817.1	1.658	0.603	0.5800	0.9385
275.3	290	414.4	389.4	1203.6	814.2	1.602	0.624	0.5840	0.9316
285.3	300	417.5	392.7	1204.1	811.3	1.551	0.645	0.5878	0.9251
295.3	310	420.5	395.9	1204.5	808.5	1.502	0.666	0.5915	0.9187
305.3	320	423.4	399.1	1204.9	805.8	1.456	0,687	9.5951	0.9125
315.3	330	426.3	402.2	1205.3	803.1	1.413	0.708	0.5986	0.9065
325.3	340	429.1	405.3	1205.7	800.4	1.372	0.729	0.6020	0.9006
335.3	350	431.9	408.2	1206.1	797.8	1.334	0.750	0.6053	0.8949
345.3	360	434.6	411.2	1206.4	795.3	1.298	0.770	0.6085	0.8894
355.3	370	437.2	414.0	1206.8	792.8	1.264	0.791	0.6116	0.8840
365.3	380	439.8	416.8	1207.1	790.3	1.231	0.812	0.6147	0.8788
375.3	390	442.3	419.5	1207.4	787.9	1.200	0.833	0.6178	0.8737
385.3	400	444.8	422	1208	786	1.17	0.86	0.621	0.868
435.3	450	456.5	435	1209	774	1.04	0.96	0.635	0.844
485.3	500	467.3	448	1210	762	0.93	1.08	0.648	0.822
535.3	550	477.3	459	1210	751	0.83	1.20	0.659	0.801
585.3	600	486.6	469	1210	741	0.76	1.32	0.670	0.783
Source	684	500	484	1209	725	0.66	1.52	0.686	0.755
	1062	550	542	1200	658	0.42	2.36	0.743	0.650
	1574	600	604	1176	572	0.27	3.75	0.799	0.540
	2265	650			441	0.16	6.2		0.396
	2974	689				0.05			
*	3075	700							
*	4300.2	752							
†	5017.1	779							
†	5659.9	810.6							

*From G. A. Goodenough's tables 1915.

†Calculated by J. McFarlane Gray—*Proc. Inst. M.E.*, July, 1889.

Properties of Superheated Steam

(Condensed from Marks and Davis' Steam Tables and Diagrams.)

v = specific volume in cu. ft. per lb., h = total heat, from water at 32° F. in B.t.u. per lb., n = entropy, from water at 32°.

Press. Abs. Lbs. per Sq. In.	Temp. Sat. Steam.	Degrees of Superheat.									
		0	50	100	150	200	250	300	400	500	600
20	228.0	v 20.08	21.69	23.25	24.80	26.33	27.85	29.37	32.39	35.40	38.40
		h 1156.2	1179.9	1203.5	1227.1	1250.6	1274.1	1297.6	1344.8	1392.2	1440.0
		n 1.7320	1.7652	1.7961	1.8251	1.8524	1.8781	1.9026	1.9479	1.9893	2.0275
40	267.3	v 10.49	11.33	12.13	12.93	13.70	14.48	15.25	16.78	18.30	19.80
		h 1169.4	1194.0	1218.4	1242.4	1266.4	1290.3	1314.1	1361.6	1409.3	1457.4
		n 1.6761	1.7089	1.7392	1.7674	1.7940	1.8189	1.8427	1.8867	1.9271	1.9646
60	292.7	v 7.17	7.75	8.30	8.84	9.36	9.89	10.41	11.43	12.45	13.46
		h 1177.0	1202.6	1227.6	1252.1	1276.4	1300.4	1324.3	1372.2	1420.0	1468.2
		n 1.6432	1.6761	1.7062	1.7342	1.7603	1.7849	1.8081	1.8511	1.8908	1.9279
80	312.0	v 5.47	5.92	6.34	6.75	7.17	7.56	7.95	8.72	9.49	10.24
		h 1182.3	1208.8	1234.3	1259.0	1283.6	1307.8	1331.9	1379.8	1427.9	1476.2
		n 1.6200	1.6532	1.6833	1.7110	1.7368	1.7612	1.7840	1.8265	1.8658	1.9025
100	327.8	v 4.43	4.79	5.14	5.47	5.80	6.12	6.44	7.07	7.69	8.31
		h 1186.3	1213.8	1239.7	1264.7	1289.4	1313.6	1337.8	1385.9	1434.1	1482.5
		n 1.6020	1.6358	1.6658	1.6933	1.7188	1.7428	1.7656	1.8079	1.8468	188.29
120	341.3	v 3.73	4.04	4.33	4.62	4.89	5.17	5.44	5.96	6.48	6.99
		h 1189.6	1217.9	1244.1	1269.3	1294.1	1318.4	1342.7	1391.0	1439.4	1487.8
		n 1.5873	1.6216	1.6517	1.6789	1.7041	1.7280	1.7505	1.7924	1.8311	1.8669
140	353.1	v 3.22	3.49	3.75	4.00	4.24	4.48	4.71	5.16	5.61	6.06
		h 1192.2	1221.4	1248.0	1273.3	1298.2	1322.6	1346.9	1395.4	1443.8	1492.4
		n 1.5747	1.6096	1.6395	1.6666	1.6916	1.7152	1.7376	1.7792	1.8177	1.8533
160	363.6	v 2.83	3.07	3.30	3.53	3.74	3.95	4.15	4.56	4.95	5.34
		h 1194.5	1224.5	1251.3	1276.8	1301.7	1326.2	1350.6	1399.3	1447.9	1496.6
		n 1.5639	1.5993	1.6292	1.6561	1.6810	1.7043	1.7266	1.7680	1.8063	1.8418
180	373.1	v 2.53	2.75	2.96	3.16	3.35	3.54	3.72	4.09	4.44	4.78
		h 1196.4	1227.2	1254.3	1279.9	1304.8	1329.5	1353.9	1402.7	1451.4	1500.3
		n 1.5543	1.5904	1.6201	1.6468	1.6716	1.6948	1.7169	1.7581	1.7962	1.8316
200	381.9	v 2.29	2.49	2.68	2.86	3.04	3.21	3.38	3.71	4.03	4.34
		h 1198.1	1229.8	1257.1	1282.6	1307.7	1332.4	1357.0	1405.9	1454.7	1503.7
		n 1.5456	1.5823	1.6120	1.6385	1.6632	1.6862	1.7082	1.7493	1.7872	1.8225
220	389.9	v 2.09	2.28	2.45	2.62	2.78	2.94	3.10	3.40	3.69	3.98
		h 1199.6	1232.2	1259.6	1285.2	1310.3	1335.1	1359.8	1408.8	1457.7	1506.8
		n 1.5379	1.5753	1.6049	1.6312	1.6558	1.6787	1.7005	1.7415	1.7792	1.8145
240	397.4	v 1.92	2.09	2.26	2.42	2.57	2.71	2.85	3.13	3.40	3.67
		h 1200.9	1234.3	1261.9	1287.6	1312.8	1337.6	1362.3	1411.5	1460.5	1509.8
		n 1.5309	1.5690	1.5985	1.6246	1.6492	1.6720	1.6937	1.7344	1.7721	1.8072
260	404.5	v 1.78	1.94	2.10	2.24	2.39	2.52	2.65	2.91	3.16	3.41
		h 1202.1	1236.4	1264.1	1289.9	1315.1	1340.0	1364.7	1414.0	1463.2	1512.5
		n 1.5244	1.5631	1.5926	1.6186	1.6430	1.6658	1.6874	1.7280	1.7655	1.8005

Properties of Superheated Steam—*Continued*

Press. Abs. Lbs. per Sq. In.	Temp. Sat. Steam.	Degrees of Superheat.									
		0	50	100	150	200	250	300	400	500	600
280	411.2	v 1.66	1,81	1.95	2.09	2.22	2.35	2.48	2.72	2.95	3.19
		h 1203.1	1238.4	1266.2	1291.9	1317.2	1342.2	1367.0	1416.4	1465.7	1515.1
		n 1.5185	1.5580	1.5873	1.6133	1.6375	1.6603	1.6818	1.7223	1.7597	1.7945
300	417.5	v 1.55	1.69	1.83	1.96	2.09	2.21	2.33	2.55	2.77	2.99
		h 1204.1	1240.3	1268.2	1294.0	1319.3	1344.3	1369.2	1418.6	1468.0	1517.6
		n 1.5129	1.5530	1.5824	1.6082	1.6323	1.6550	1.6765	1.7168	1.7541	1.7889
350	431.9	v 1.33	1.46	1.58	1.70	1.81	1.92	2.02	2.22	2.41	2.60
		h 1206.1	1244.6	1272.7	1298.7	1324.1	1349.3	1374.3	1424.0	1473.7	1523.5
		n 1.5002	1.5423	1.5715	1.5971	1.6210	1.6436	1.6650	1.7052	1.7422	1.7767
400	444.8	v 1.17	1.28	1.40	1.50	1.60	1.70	1.79	1.97	2.14	2.30
		h 1207.7	1248.6	1276.9	1303.0	1328.6	1353.9	1379.1	1429.0	1478.9	1528.9
		n 1.4894	1.5336	1.5625	1.5880	1.6117	1.6342	1.6554	1.6955	1.7323	1.7666
450	456.5	v 1.04	1.14	1.25	1.35	1.44	1.53	1.61	1.77	1.93	2.07
		h 1209	1252	1281	1307	1333	1358	1383	1434	1484	1534.0
		n 1.479	1.526	1.554	1.580	1.603	1.626	1.647	1.687	1.723	1.758
500	467.3	v 0.93	1.03	1.13	1.22	1.31	1.39	1.47	1.62	1.76	1.89
		h 1210	1256	1285	1311	1337	1362	1388	1438	1489	1539
		n 1.470	1.519	1.548	1.573	1.597	1.619	1.640	1.679	1.715	1.750

Volume of Superheated Steam—Linde's equation (1905),

$$pv = 0.5962T - p(1 + 0.0014p)\left(\frac{150{,}300{,}000}{T^3} - 0.0833\right)$$

in which p, is in lb. per sq. in., v, is in cu. ft. and T, is the absolute temperature on the Fahrenheit scale, has been used in the computation of Marks & Davis' steam tables.

Specific heat of superheated steam.—Mean specific heats from the temperature of saturation to various temperatures at several pressures—*Knoblauch and Jakob* (from Peabody's Tables).

Lb. per sq. in.		14.2	28.4	56.9	85.3	113.3	142.2	170.6	199.1	227.5	256.0	284.4
Temp. sat. °F.		210	248	289	316	336	350	368	381	392	403	412
°F.	°C.											
212	100	0.463										
302	150	.462	0.478	0.515								
392	200	.462	.475	.502	0.530	0.560	0.597	0.635	0.677			
482	250	.463	.474	.495	.514	.532	.552	.570	.588	0.609	0.635	0.664
572	300	.464	.475	.492	.505	.517	.530	.541	.550	.561	.572	.585
662	350	.468	.477	.492	.503	.512	.522	.529	.536	.543	.550	.557
752	400	.473	.481	.494	.504	.512	.520	.526	.531	.537	.542	.547

CHAPTER 2

THE STEAM ENGINE

Ques. What is a steam engine?

Ans. *A machine for converting heat into mechanical power.*

Ques. Into what three classes are engines divided with respect to service?

Ans. Stationary, marine and locomotive.

Ques. Into what two classes are engines divided with respect to their mode of operation?

Ans. Non-condensing and condensing.

Ques What is a non-condensing engine?

Ans. A non-condensing engine (sometimes called a simple, or high pressure engine) is one that exhausts against the pressure of the atmosphere.

Ques. What is a condensing engine?

Ans. A condensing engine is one that exhausts into a *condenser* or device which condenses the exhaust steam, and in which, by means of an air pump, a partial vacuum is maintained, thus reducing the back pressure.

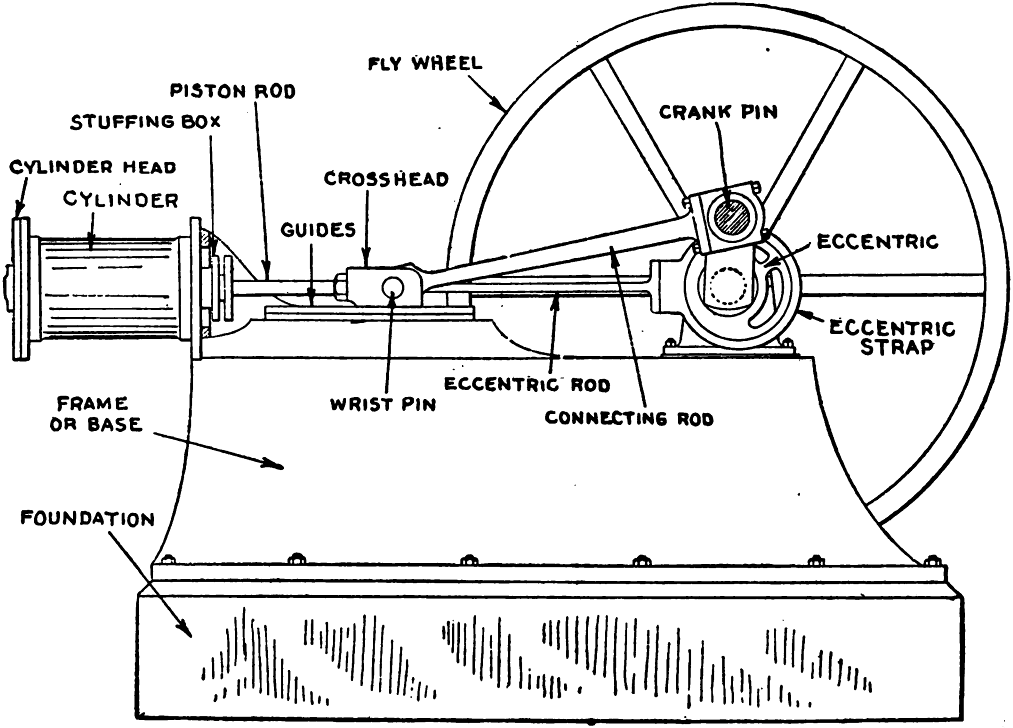

FIG. 53.—Elevation showing parts of a horizontal steam engine. *As shown,* the engine is mounted on a substantial frame or base, to which is bolted at one end, the main bearing, and at the other, the cylinder, being connected by a vertical flange projecting from the top of the frame. The figure shows the general arrangement of parts.

How an Engine Works.—In a steam engine, heat accomplishes work only by being "let down" from a *higher* to a *lower temperature;* in the process some of the heat is converted into useful work. The mechanism by which this is accomplished is not so complicated as would at first seem, and its operation is easily understood.

Fig. 54 is a sectional plan view of a simple form of steam engine. C, is a cylinder into which steam is admitted *alternately* by the valve V, through the steam passages S, S'. This causes a steam tight piston P, to move back and forth in the cylinder.

The pressure of the steam on the piston is transmitted through a *piston rod* to a *connecting rod* CR, which causes the *crank* K, to revolve; thus, the *reciprocating motion* of *the piston* is transformed into *rotary motion* of the crank.*

In the revolution of the crank, the connecting rod will make various angles with the piston rod, hence, to allow for this, a *cross head* H, is placed at the point where the two rods meet, thus forming a hinged joint. The cross head is provided with *guides* to prevent the piston rod being broken or bent by the oblique thrusts and pulls which it imparts to the crank by means of the connecting rod. The crank is keyed or forged to a shaft Z, upon which is fastened a *fly wheel.*

In the operation of the engine, it is evident that while steam is being admitted at one end of the cylinder, the supply already in the cylinder from the previous stroke, must be exhausted from the other end. This is accomplished by means of a slide valve V.

The two steam passages S, S', connect the ends of the cylinder with a box-like projection M, called the *steam chest.* These passages terminate in a smooth flat surface V S, known as the *valve seat,* and upon which the valve moves; the ***ends*** of the passages terminating at the valve seat being called the *ports.* Careful distinction should be made between the terms *passages* and *ports.*

The two ports just mentioned are called the *steam ports,* to distinguish them from a third and larger port located midway between them and

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*NOTE.—When ***James Watt*** produced his ***"rotative engine"*** in 1780 he was unable to use the crank because it had already been patented by Matthew Wasborough. Watt was not discouraged and within one year had himself patented five other devices for obtaining rotary motion from a piston rod.

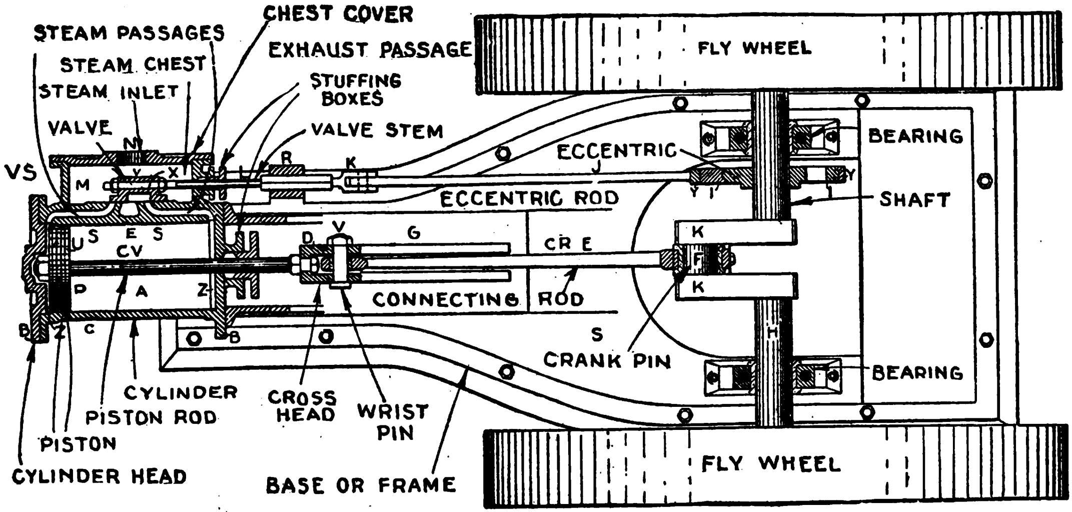


FIG. 54.—Plan showing parts of a horizontal steam engine, with names of same. The numerous parts may be classified as: *1, stationary; 2, reciprocating; 3, rotating.* Thus, *1,* the frame, cylinder, cylinder heads, valve chest cover, stuffing box guides, and main bearing are *stationary parts:* **2,** the piston, piston rod, cross head, wrist pin, valve, valve stem, eccentric rod, eccentric strap are *reciprocating parts;* **3,** the fly wheel, shaft, eccentric, are *rotating parts.* The engine here shown may be classed as a simple slide valve horizontal engine.

called the *exhaust port,* through which steam passes from the cylinder to the *exhaust pipe.* The transverse form of these passages is long and narrow, so that steam may be quickly admitted and exhausted from the engine with only a slight valve movement.

The valve itself, is a rectangular iron box, having a cavity, similar in form to the letter D. The size of the valve is so proportioned that in moving back and forth over the valve seat it will alternately cover and uncover the two steam ports, allowing steam to flow alternately into the cylinder ends from the steam chest.

The exhaust cavity EC, connects either steam port to the exhaust port, so that while steam is being admitted to one end of the cylinder it is exhausted from the other end.

The mechanism which imparts the to and fro motion to the valve is called the *valve gear,* and is quite similar to the connections between the piston and crank.

Instead of a crank, there is usually an *eccentric* to impart motion to the valve. This consists of a disc bored out of the center and fastened by a key, or set screw to the shaft. Around the eccentric is a grooved ring called *the eccentric strap.* Motion is transmitted from the eccentric to the valve by means of an *eccentric rod* and *valve stem* as shown, a *valve stem guide* being provided to prevent the valve stem springing out of position on account of the side thrust of the eccentric rod. By turning the eccentric on the shaft, the relation between the valve movement and piston movement may be changed, hence, the eccentric may be adjusted so as to give the proper distribution of steam to and from the cylinder.

In the figure, the piston is shown at the end of the cylinder or just beginning the stroke. In this position, the piston rod and eccentric rod are in a straight line, so that no matter how much steam pressure there may be on the piston it will not cause the crank to rotate. This happens when the piston is at either end of the cylinder, the corresponding positions of the crank pin being called *dead centers.*

To prevent the engine stopping on a dead center, a *fly wheel* having a heavy rim is provided, which by its *momentum* keeps the engine in motion in passing these centers.

Ques. Describe the operation of the engine.

Ans. As shown in fig. 54, the piston is at the beginning of the stroke and the valve has just begun to open the steam port, admitting steam to the cylinder. As the piston moves, the valve opens the port to its full extent and closes it before the stroke is completed, thus "cutting off" the supply of steam. During these "events" of the power stroke the exhaust cavity of the valve connects the other steam port with the exhaust port E allowing the steam which was admitted during the previous stroke to exhaust into the atmosphere, or condenser, according to whether the engine be run non-condensing or condensing.

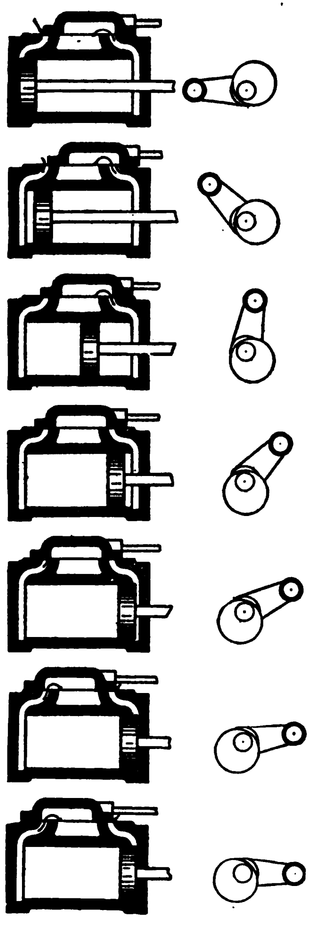
Since the supply of steam is cut off by the valve before the piston completes the stroke, the steam in the cylinder expands from the point of cut off until the piston has almost completed the stroke. It is released at this point by the valve connecting the proper steam port with the exhaust port E.

The movement of the valve, eccentric and piston during one revolution of the crank may be more easily understood by the aid of a series of diagrams, figs. 55 to 61. The crank and eccentric positions corresponding to the several piston and valve positions, are shown at the right.

In each figure the center of the crank pin, center of the shaft and the eccentric are shown at the right.

In fig. 55 the piston is at the beginning of the stroke; the valve has just begun to open the steam port to the left for the admission of steam, while the steam port to the right is fully open for exhaust. To bring the valve in this position, the eccentric has been set *in advance* of the crank as indicated. The reason for this will be explained later.

In fig. 56, the piston has advanced to the right about $^{1}/_{10}$ of the stroke and the valve has moved so that the port at the left is nearly wide open for the



FIGS. 55 to 61.—Diagrams showing several positions of the piston, valve, crank and eccentric during one stroke. The diagrams show the relative movements of the parts, the crank and eccentric positions being shown at the right.

admission of steam. Up to this point the piston and valve have been moving in the same direction.

The valve now begins its return stroke and when the piston has moved about $^{6}/_{10}$ of its stroke, the valve has just closed the port at the left, thus cutting off the steam supply as shown in fig. 57. The steam now expands as the piston continues to move, but during this interval the port to the right is gradually being closed to the exhaust.

When the piston has moved about $^{9}/_{10}$ of its stroke, this port is closed and the steam remaining in the cylinder at that end is *compressed* which helps to bring the piston to rest without jar as it reaches the end of the stroke.

The velocity of the piston is greatly reduced as it nears the end of the stroke while the movement of the valve is increased.

In fig. 59 when the piston has moved only slightly from its position in the preceding figure, the valve is at the point of opening the port at the left to *release* the steam for exhaust.

In the next two figures the piston completes its stroke, while the port at the left is being very rapidly opened to exhaust.

Fig. 61 shows the piston at the end of the stroke and the valve just beginning to admit steam to the port at the right for the return stroke, completing one stroke, the same cycle of events being repeated for the return stroke.

The Expansion of Steam.—In the operation of a steam engine, steam, as just explained, is admitted and exhausted alternately at the ends of a cylinder within which is a piston. The force exerted by the steam causes the piston to move to and fro which by suitable connections is made to do useful work. The distance the piston moves in either direction is called the *stroke*.

*When engines are required to exert their full power for a short period as happens, for instance, when a locomotive is pulling a heavy train up an incline, steam is admitted to the cylinder at full pressure through the greater

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*NOTE.—Steam was first used *expansively* in an engine by Watt who in making application for a patent said, "My improvement in steam engines consists in admitting steam into the cylinder, or steam vessels of the engine only during some part or portion of the descent or ascent of the piston of said cylinder, and using the elastic forces, wherewith the said steam expands itself in proceeding to occupy larger spaces as the acting powers on the piston through the other parts or portions of the length of the stroke of said piston." All engines now operate on this principle except where extraordinary conditions prevail.

part of each stroke, without regard to economy in its use. However, ***this is* not *the way the medium is ordinarily used in an engine***, for although an extra amount of work is done, it is at the expense of an excessive proportion of steam and fuel compared with the gain in work.

Boyle's Law.—The behavior of a gas in expanding has been stated by Boyle as follows: *The pressure of a perfect gas at constant temperature varies inversely as its volume.**

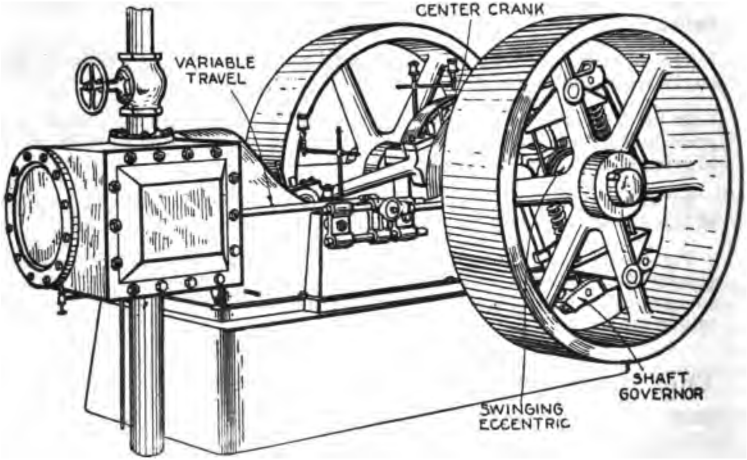

Fig. 62.—The Atlas medium speed center crank engine with automatic cut off. This engine adjusts its power output to meet fluctuations in the load by the automatic action of the shaft governor which varies the point of cut off. Steam is expanded to a higher degree than with a throttling engine resulting in superior economy.

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*NOTE.—The student should distinguish between ***isothermal*** and ***adiabatic*** expansion. Isothermal expansion means ***expansion at constant temperature;*** adiabatic expansion denotes expansion ***without receiving or giving up heat.*** It should be noted that the expansion of steam in an engine cylinder is ***neither isothermal nor adiabatic.*** According to Rankin, when steam expands in a closed cylinder, as in an engine, the approximate law of the expansion is $P \propto V^{-10/9}$, or $PV^{1.111}$ = a constant. The curve constructed from this formula is called the *adiabatic* curve. The author does not believe the expansion of steam even approximately follows the adiabatic curve, and may be said to depart considerably therefrom, especially with very early cut off where the effects of condensation and re-evaporation are marked. Peabody says: "It is probable that this equation (Rankin's equation above mentioned) was obtained by comparing the expansion lines on a large number of indicator diagrams." He states also that "there does not appear to be any good reason for using an exponential equation in this connection." Also that "the action of a lagged steam engine cylinder ***is far from being adiabatic.***" **For general calculation** ***steam may be taken as expanding in the cylinder according to Boyle's law, above given.***

This law may be illustrated by the following experiment: In fig. 63 is shown a cylinder, having a piston sliding air tight in its length. If air be compressed in front of the piston as it is forced from one end toward the other, the pressure exerted by the air will increase in ratio as the volume is diminished. This fact may be shown by inserting in the wall of the cylinder at different points a number of tubes, each provided with an air tight piston upon which bears a spiral spring holding it, as at A, when the pressure on the piston is the same on both sides.

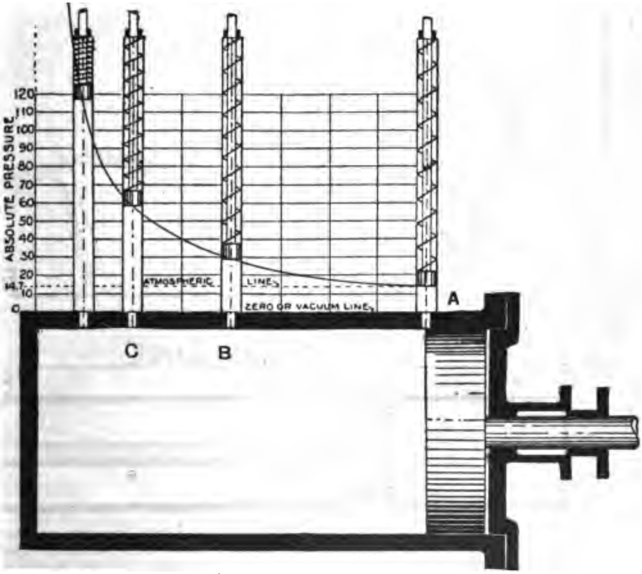

FIG. 63.—Experiment illustrating Boyle's law. This law was discovered by the Hon. Robert Boyle, in 1660, and in 1661 he presented to the Royal Society his work, "Touching the Spring of Air and its Effects." With respect to the experiment on air he says: "'Tis evident that as common air when reduced to half its natural extent obtained a spring about twice as forcible as it had before, so the air, being thus compressed; being further crowded into half this narrow space, obtained a spring as strong again as that it last had, and consequently four times as strong as that of common air." Boyle does not appear to have considered his law to possess the wide application afterwards credited to it. He believed that for pressure above four atmospheres, the compression of air was less than the amount corresponding to the law.

The area of each small piston is assumed to be one square inch, and the spring of such a tension that it will move upward through one of the spaces between the horizontal lines on the diagram with each ten pounds of added pressure in the large cylinder.

Now when the piston moves in the cylinder, the pressure will gradually rise due to the compression of the air and the small pistons will rise against the tension of the springs to increasing heights.

As the piston moves from the end of the cylinder to the following points:

initial position, ½ stroke ¾ stroke ⅞ stroke

the positions of the small pistons as shown in the figure will indicate the following pressures:

14.7 lbs. 29.4 lbs. 58.8 lbs. 117.6 lbs.

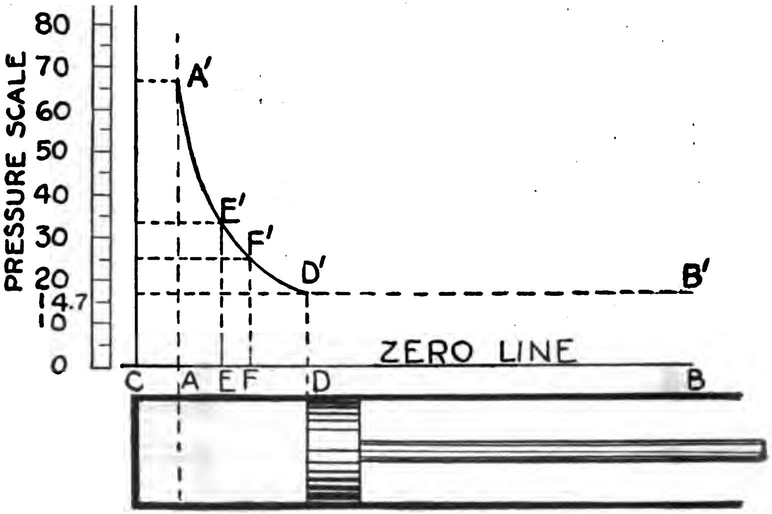

FIG. 64.—To describe the hyperbolic curve for compression. Draw the zero or vacuum line A B, (any convenient scale) = length of stroke or volume displaced by the piston, and extend it to C, making A C, of length = clearance, that is, if the clearance volume be say, 8 per cent of piston displacement then length of A C = 8% of A B, or $.08 \times$ A B. If the engine be running non-condensing and exhausting at say 2 lbs. gauge pressure, that is $14.7 + 2 = 16.7$ lbs. absolute, this would be represented by a horizontal line at a height above A B, corresponding to 16.7 lbs., on the scale of pressures, as the dotted line beginning at B′. Suppose the exhaust valve to close when the piston has reached the point D, corresponding to D′ in the diagram, then C D, represents the volume to be compressed. **By Boyle's law,** ***the pressure is inversely proportional to the volume,*** hence, when the piston has moved to E, reducing the volume, one-half, the pressure will be doubled and equal to $16.7 \times 2 = 33.4$ lbs. Measuring up from E a length corresponding to 33.4 lbs. gives E′, a point on the curve. Similarly, when the piston moves to A, compressing to one-quarter the original volume C D, the pressure rises to $16.7 \times 4 = 66.8$ lbs., giving the point A′, on the curve. The curve may be described through the points D′, E′, A′, just obtained, or if greater accuracy be desired more points may be obtained, thus, **by Boyle's law,** ***pressure × volume = constant,*** from which, ***pressure = constant ÷ volume.*** If C D = say, 3 ins., then the constant $= 3 \times 16.7 = 50.1$, hence, when the piston has moved to any point as F, reducing the volume to 2, then pressure for position F, $= 50.1 \div 2 = 25.05$ lbs. abs. Similarly, other points may be obtained, but it should be noted that it is a waste of time to obtain more than say 4 points.

thus showing that if the volume be diminished by half, the pressure is doubled. If a curve be drawn so as to pass through the center of each of the small pistons, it will show the pressure corresponding to every position of the large piston.

To apply this to the conditions of operation in a steam engine, it may be assumed that the piston has moved from the left end of the cylinder to a point C, or ¼ stroke, and during this time, steam is admitted at a constant pressure of 58.8 lbs. and then the supply cut off.

If the piston now move to B, the steam will *expand* to double its volume and its pressure will be reduced to half, or 29.4 lbs.

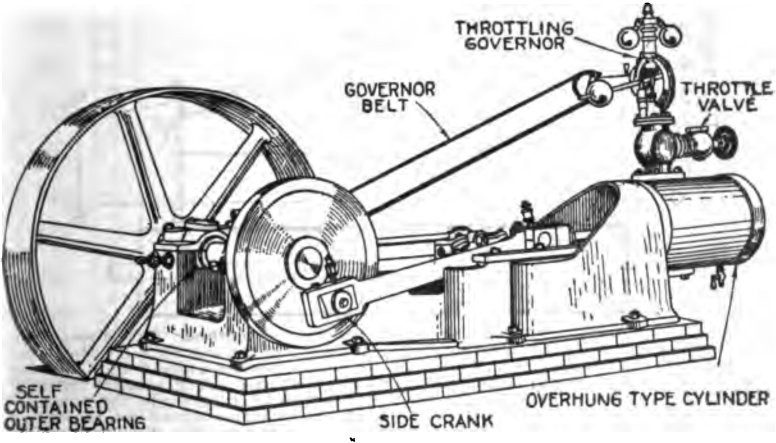

Fig. 65.—The Houston, Stanwood, and Gamble, side crank self-contained, throttling engine. The steam supply is "throttled" or varied automatically by the governor to meet load changes, the point of cut off being the same for all loads. This type of engine is cheaper than an automatic cut off engine but is not as economical in the use of steam.

Again, if the piston move to the end of the stroke (to A) the volume thus obtained would be four times the original volume, and the pressure one-fourth the original pressure, or 14.7 lbs.

The curve shows the expansion of the steam for any position of the piston during the expansion and is, therefore, called the *curve of expansion.**

Ques. What curve is taken ordinarily to represent the expansion of steam in an engine?

Ans. The equilateral or rectangular hyperbola referred to its asymptotes.*

The Saving due to Expansion.—The advantage of using steam expansively may be clearly shown by a diagram as in fig. 71, which represents the work done by an engine during

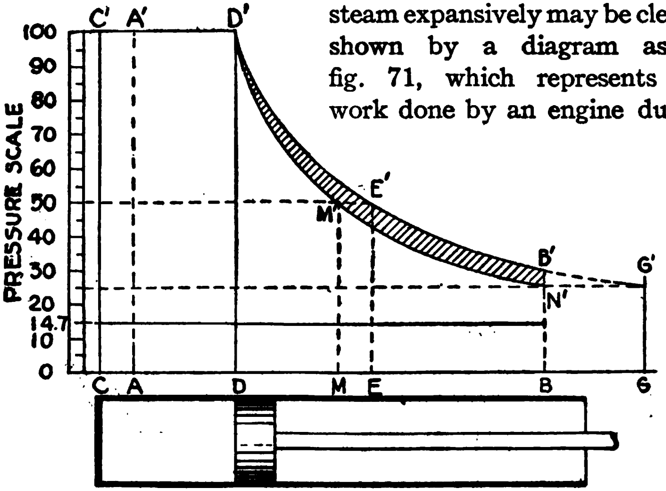

FIG. 66.—To describe the hyberbolic curve for expansion, **1st method**: This is practically the reverse of the process described in fig. 64. As before, draw the zero line A B = stroke and volume displaced by the piston. When clearance is to be considered extend it to C, making A C = clearance, as explained in fig. 64. At D, piston position at cut off, measure vertically a distance = pressure at cut off giving the point D′, then C′ D′, represents the volume of steam admitted considering clearance, and A′ D′, the volume admitted not considering clearance. **Applying Boyle's law, 1,** *considering clearance*, when the initial volume C D, is doubled by expansion to point E, pressure is reduced to ½, that is E E′ = ½ D D′. Similarly G G′ = ½ E E′, or ¼ D D′, giving the points D′, E′, G′, through which the curve passes. 2, *If clearance be not considered*, as is often the case, the initial volume is taken as A D. Again applying Boyle's law, points D′ M′ N′ are obtained giving the lower curve. Considering a theoretical diagram the error introduced by not considering clearance is here indicated by the shaded area D′ E′ B′ N′ M′ D′, however, such error is usually allowed for in fixing the value of the diagram factor as explained in the accompanying text.

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*NOTE.—The hyperbolic curve is of such importance that it should be thoroughly understood. The following definitions, accordingly, should be carefully noted: **Hyperbola.**—*A plane curve such that the difference of the distances from any point on it to two fixed points, called the* **foci**, *is equal to a given distance.* The line *passing through* the foci and terminating at the two branches of the curve is the *transverse axis*, and a line perpendicular to this axis drawn half way between the foci is the *conjugate axis*. An asymptote of a hyperbola is *a right line which an infinite branch of the curve continually approaches but does not reach, in other words, a tangent to the curve at infinity.* **The equilateral hyperbola.**—*A hyperbola whose asymptotes are perpendicular to each other.* This is the form of hyperbola which represents the law of expansion of steam, or Boyle's law. In this hyperbola, the product of the abscissa and ordinate at any point is equal to the product of abscissa and ordinate of any other point, that is, if p, be the ordinate at any point and v, its abscissa and p' and v', are the ordinate and abscissa at any other point, then $pv = p'v'$, or $p\,v$ = a constant. See Boyle's law, page 54. **Abscissæ and** one stroke. As in the preceding figure the vertical distances represent pressures and horizontal distances, piston positions.

In the figure, F D, is the length of the stroke; the line being placed at such a height above a horizontal line of no pressure, that any point on it is at atmospheric pressure. This line, therefore, is called the *atmospheric line* in distinction from the line of no pressure, or *vacuum line*.

The distance between these lines depends on the *scale* used in measuring pressures, thus, if $^1/_{10}$ inch of vertical distance be taken to represent one pound pressure, the distance between the lines will be $.1 \times 14.7 = 1.47$ inch.

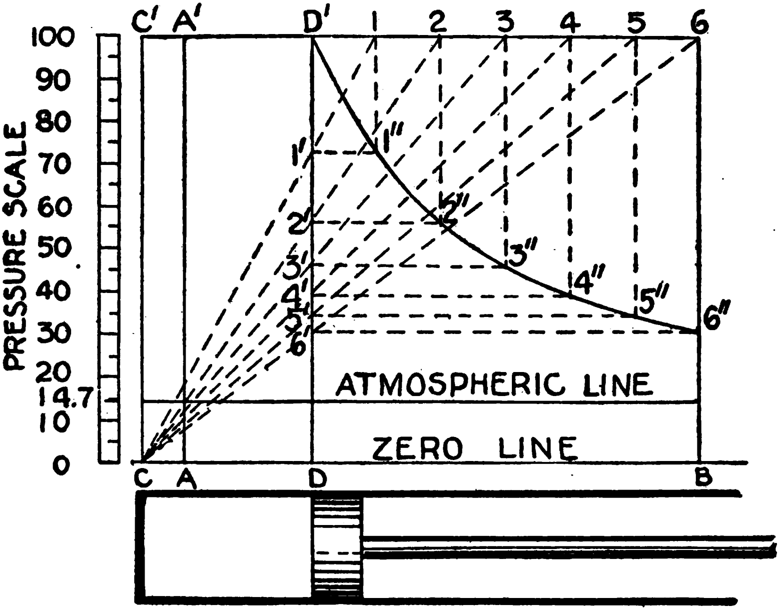

FIG. 67.—To describe the hyperbolic curve for expansion, **2nd method:** Draw the zero line A B = stroke, and extend it to C, making A C = clearance. Take the distance C C′ = admission pressure, and at the point of cut off D, erect the perpendicular D D′, giving the admission line A′ D′. Extend C′ D′, and lay off any number of points 1, 2, 3, etc. From C, draw radial lines, C1, C2, C3, etc. Draw horizontal lines through 1′, 2′, 3′, etc., and vertical lines through 1, 2, 3, etc. Their intersections 1″, 2″, 3″, are points on the hyperbolic curve.

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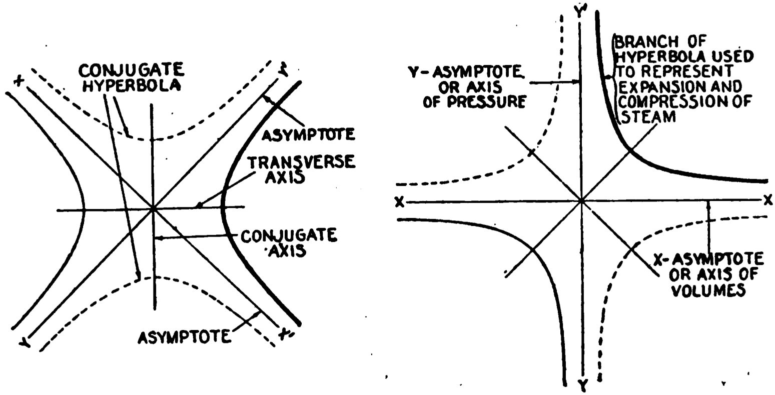
*NOTE.—*Continued.*

ordinates.—If X and Y, be two intersecting axes, X, being the axis of abscissæ, and, Y, the axis of ordinates, then, the distance of any point P, from the Y axis, measured parallel to the axis of X′, is called the abscissa of the point; also, the distance from the X axis, measured parallel to the Y axis is called the *ordinate*. The abscissa and ordinate taken together are called the co-ordinates of the point P.


CONJUGATE HYPERBOLA
ASYMPTOTE
TRANSVERSE AXIS
CONJUGATE AXIS
ASYMPTOTE
Y'
Y-ASYMPTOTE OR AXIS OF PRESSURE
BRANCH OF HYPERBOLA USED TO REPRESENT EXPANSION AND COMPRESSION OF STEAM
X
X
X-ASYMPTOTE OR AXIS OF VOLUMES
Y

Figs. 68 and 69.—Appearance of equilateral hyperbola 1, ***as referred to its rectangular axes,*** **fig. 68, and 2,** ***as referred to its rectangular asymptotes,*** **fig. 69, one branch of the hyperbola in this position being used to represent the expansive action of steam. Comparing the two figures it will be noted that fig. 69 is the same as fig. 68, rotated through 45 degrees, the general method of constructing the hyperbole in fig. 69, is shown in fig. 70, and other methods in the accompanying diagrams.**

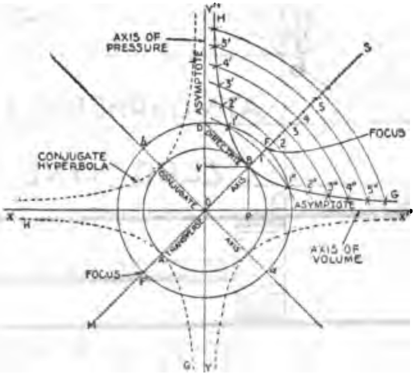


Fig. 70.—To describe an equilateral or rectangular hyperbola referred to its rectangular asymptotes. **General method:** Draw the axis of volumes, or horizontal asymptote X X′, and the axis of pressures, or vertical asymptote Y Y′, cutting X X′, at O, or hyperbolic center. Through O, draw M S, at 45° to X X′. Take any point on M S, as B, and with radius O B, describe a circle, cutting M S, in B and A, giving A B, the transverse axis. At B, erect a perpendicular cutting Y Y′, at D, giving O D, the directrix. With O D, as radius describe a circle cutting M S, at F and F′; these points are the *foci* of the hyperbola. On M S, take any number of points 1, 2, 3, etc., and from F, and F′ as centers, with A1, B1, A2, B2, etc., as radii, describe arcs cutting each other in 1′, 2′, 3′, etc., and 1″, 2″, 3″, etc., through which points the branch H B G, of the hyperbola is described. Similarly, the other branch H A G,

If steam be admitted at say, 100 lbs. per square inch ***absolute pressure*** when the piston is at F, and the supply continued at constant pressure

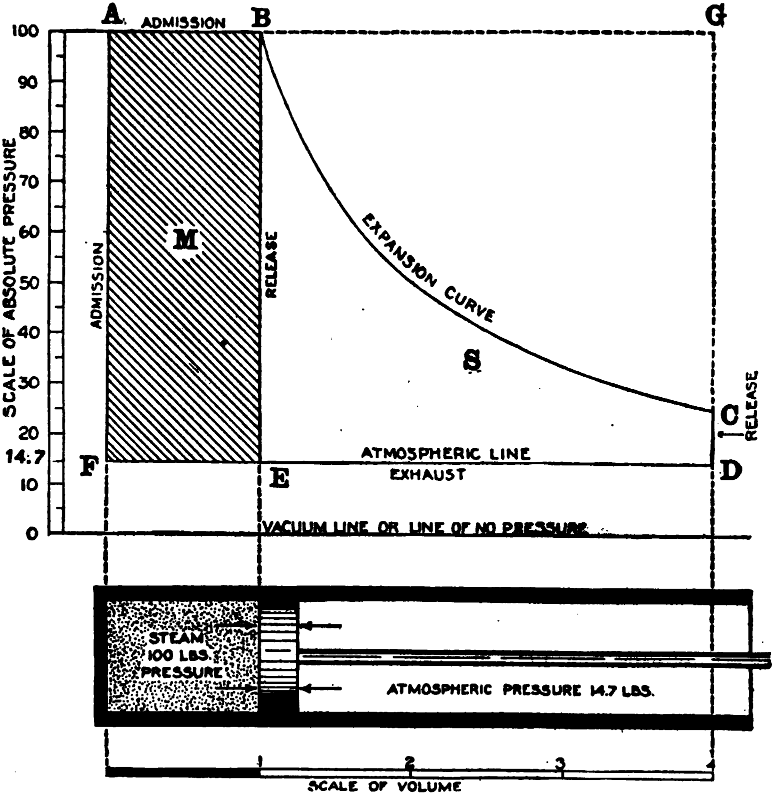


Fig. 71.—Theoretical card or diagram showing the theoretical advantage of using steam expansively.

Fig. 70.—*Continued.*

and conjugate hyperbola shown in dotted lines may be described, but for the purpose in view, only one branch H B 6 need be described. It is a property of the hyperbola referred to its rectangular asymptotes, as above, that if one asymptote as X X′, be taken as an axis of volumes and the other an axis of pressures measured from the intersection O, then for any point on the curve as B, the product of its distance from Y Y′, multiplied by its distance from X X′, = constant, that is B V × B P = constant, or pressure × volume = constant which is in accordance with Boyle's law.

for one-quarter of the stroke, namely, till the piston reaches E, as shown, this may be represented in the diagram by the horizontal line AB, drawn at a height corresponding to 100 lbs. pressure.

Now, if the steam be *released*, the pressure will fall to that of the atmosphere as indicated by the line BE, and the work done *for each square inch of piston area*, will be equal to the *area* of the shaded rectangle ABEF, or M, because

$$\textit{work done} = \overset{(\textit{load})}{AF} \times \overset{(\textit{distance moved})}{FE} = \overset{(\textit{area})}{ABFE}$$

If, instead of exhausting the steam when the piston has made only one-quarter stroke, the supply previously admitted be retained in the cylinder without any additional supply, and be allowed to expand, the curve of expansion BC will represent the gradual fall of pressure during the expansion.

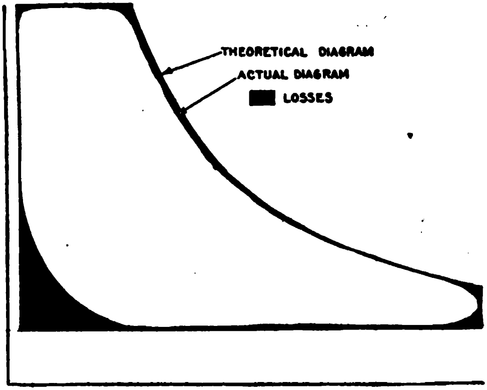


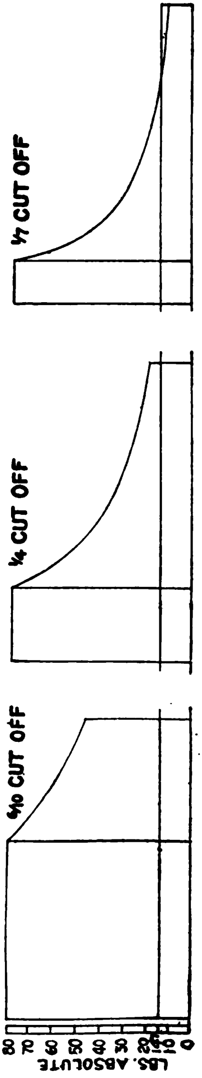
FIG. 72.—Comparison of theoretical and actual diagrams illustrated by solid black section losses in the actual engine which reduce the theoretical gain due to expansion.

At the end of the stroke, the absolute pressure C, due to the expansion is called the *terminal pressure.* At this point, the steam is *released* and its pressure falls to that of the atmosphere and then it is exhausted as indicated by the line CD.

The gain due to expanding the steam will be clearly seen by noting the size of the four sided figure BCDE, or S, as compared with the shaded figure M. This area S, represents *the work done by the steam in expanding* just as the shaded area M, represents *the work of the steam during admission,* the exact amount of gain being determined by measuring the areas of M and S, and dividing the combined area of the two by the area of M, that is,

$$\textit{gain by expansion} = \frac{M+S}{M}$$

The method of measuring these areas will be explained later.



FIGS. 73 to 75.—Diagrams illustrating various cut offs. Fig. 73, six-tenths cut off, standard for marine engines; fig. 74, one-fourth cut off, ordinarily the most economical for stationary non-condensing engines; fig. 75 one-seventh, cut off, ordinarily most economical for stationary condensing engines. In the diagrams the compression curves are not shown.

To illustrate the great waste which results in admitting steam to the cylinder the full length of stroke, as in the case of the ordinary steam pump, it may be assumed that instead of cutting off the supply at B, it be continued at the same pressure to the end of the stroke as represented by the dotted line BG.

If now the steam be released, its pressure will fall to that of the atmosphere as indicated by the line GD, and the work done during the stroke will be represented by the area of the rectangle AGDF.

The increase in power secured by admitting steam for full stroke instead of cutting off at one-quarter stroke and expanding, is indicated by the area of the three sided figure BGC. To obtain this increase in power, it should be noted, requires four times the amount of steam as when cutting off at one-quarter stroke and expanding.

By comparing the areas of the figure, it will be seen that in admitting steam for full stroke, the steam consumption is excessive, and all out of proportion to the gain in power.

Cut Off.—When steam is used expansively in a cylinder, it is admitted during a portion of the stroke at a constant pressure, and then the supply suddenly discontinued. That point of the stroke in which this occurs is called the *cut off,** and is usually expressed as a fraction of the stroke, thus ½, ⅓, etc.

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*NOTE.—This is the ***apparent cut off*** as distinguished from the ***real cut off***, later explained.

Number of Expansions.—The degree in which steam is expanded is expressed *in terms of the original volume*, thus, four expansions mean that steam has been expanded to a volume four times as large as its original volume. The number of expansions is determined by the cut off.

Rule 1. *Number of expansions equal one divided by the cut off.*

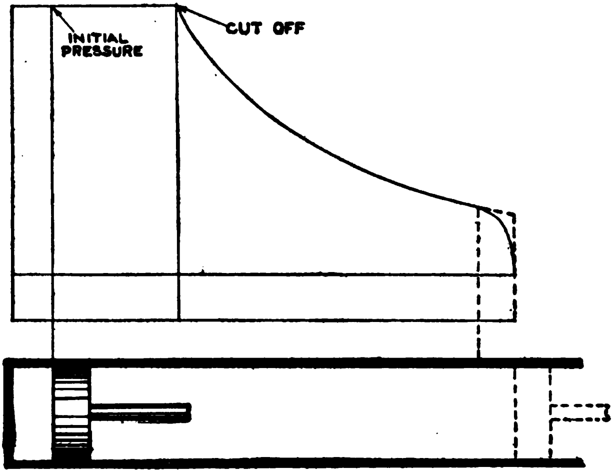

Fig. 76.—Diagram illustrating *initial pressure. This* is the *pressure in the cylinder at the beginning of the stroke and on the theoretical diagram is assumed to remain constant up to the point of cut off. Some initial pressures:* Atmospheric engines, 0 lbs. gauge; low pressure engines, 20 lbs.; early walking beam marine engines, 25 lbs.; later types 50 to 75 lbs.; stationary engines, 50 to 250 lbs.; marine screw engines, 80 to 250; locomotives,150 to 275: locomobiles and special engines, 250 to 500 lbs.

Thus, if steam be cut off at one-quarter stroke,

number of expansions $= 1 \div \frac{1}{4} = 1 \times \frac{4}{1} = 4.$

Rule 2.—*Number of expansions equal absolute pressure at cut off divided by terminal pressure.*

Thus if steam be expanded from 100 lbs. absolute cut off pressure to 20 lbs. absolute terminal pressure, number of expansion $= 100 \div 20 = 5.$

Initial Pressure.—This is *the pressure at which steam is admitted to the cylinder*, and should not be confused with the boiler pressure.* It is theoretically the same as the cut off pressure, but in practice may be quite different.

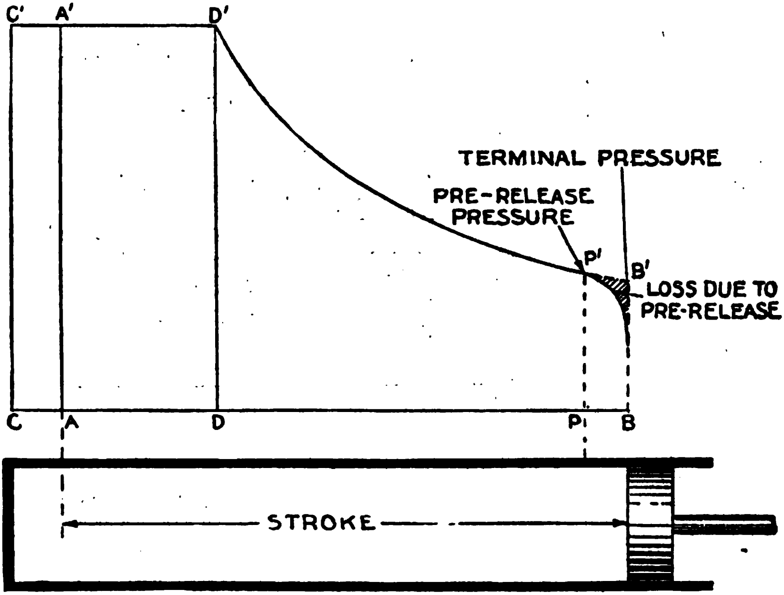

FIG. 77.—Diagram illustrating *terminal pressure*. The valve gear of a steam engine is so constructed that exhaust begins *before* the piston has completed the stroke, that is, the steam is *pre-released* when the piston is near the end of the stroke, so that (especially in the case of high speed engines) the pressure of the steam in the cylinder will be reduced as near as possible to the exhaust pressure at the beginning of the exhaust stroke. Theoretical calculations, however, are simplified by assuming that exhaust does not begin till the end of the stroke. Accordingly the pressure at that point due to expansion, called terminal pressure, may be defined as the ***imaginary pressure that would exist in the cylinder at the end of the stroke if the steam were expanded to this point instead of being pre-released.*** ***Some terminal pressures:*** Single cylinder non-condensing 25 to 20 lbs. abs., condensing 20 to 12; multi-cylinder condensing 12 to 5 lbs. abs.

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*NOTE.—In practice the initial pressure is always *less* than the boiler pressure because of the resistance offered to the flow of steam through the steam pipe, engine ports and passages, especially where the engine is at some distance from the boiler and the steam line contains numerous elbows: these conditions and condensation all contribute to cause *drop* in pressure between boiler and engine. In ordinary plants this drop is usually two or more pounds. It should be noted in applying Boyle's law that **absolute** *pressures* should be used. Thus, 90 lb. gauge boiler pressure with 2 lb. drop would give 90 + 14.7 — 2 = 102.7 lbs. **absolute** initial pressure.

Ques. In theoretical calculations what assumption is made in regard to the initial pressure?

Ans. It is assumed to remain constant during admission, that is to the point of cut off.

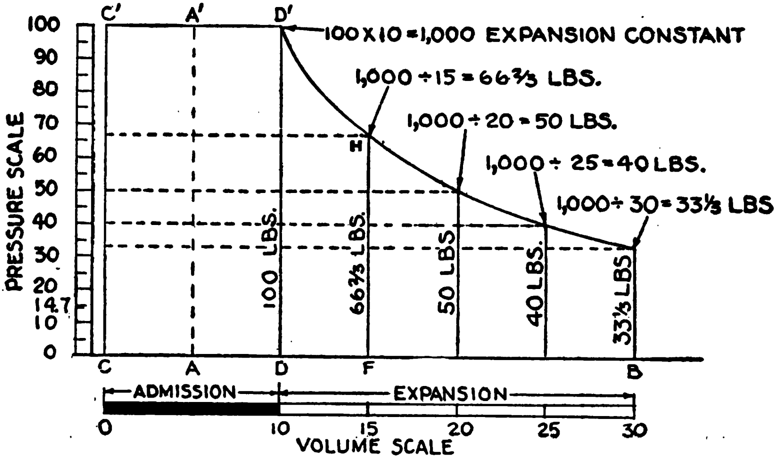


FIG. 78.—Diagram illustrating the **expansion constant** *and its use.* According to **Boyle's law,** *pressure × volume = constant.* If, as indicated in the diagram, steam be admitted to a cylinder during 10 inches of the stroke and expanded to 30 inches, the expansion constant = 100 × 10 = 1,000, from which the pressure at any other point = *constant ÷ volume*, that is, when the piston is at
15 ins. 20 ins. 25 ins. 30 ins.
of the stroke, the *expansion constant ÷ volume* is
1,000 ÷ 15 1,000 ÷ 20 1,000 ÷ 25 1,000 ÷ 30
which is equal to
66⅔ lbs. 50 lbs. 40 lbs. 33⅓ lbs.
Similarly *volume = constant ÷ pressure*, that is, when the pressure due to the expansion is
66⅔ lbs. 50 lbs. 40 lbs. 33⅓ lbs.
the *expansion constant ÷ pressure* is
1,000 ÷ 66⅔ 1,000 ÷ 50 1,000 ÷ 40 1,000 ÷ 33⅓
which is equal to
15 ins. 20 ins. 25 ins. 30 ins.

Terminal Pressure.—*If steam be expanded to the end of the stroke, the pressure at that point is called the* **terminal pressure.** It is determined from the initial pressure and the number of expansions.

Rule. ***The terminal pressure equals the initial pressure divided by the number of expansions.***

Thus, if the initial pressure be 100 lbs. absolute, and the number of expansions 4,

$$\text{terminal pressure} = 100 \div 4 = 25 \text{ lbs. absolute.}$$

Example.—If the initial pressure be 100 lbs. gauge, and the number of expansions be 4, what is the terminal gauge pressure?

$$100 \text{ lbs. gauge} = 100 + 14.7 = 114.7 \text{ lbs. absolute.}$$

$$114.7 \div 4 \begin{cases} = 28.69 \text{ lbs. absolute, or} \\ = 28.69 - 14.7 = 13.99 \text{ gauge.} \end{cases}$$

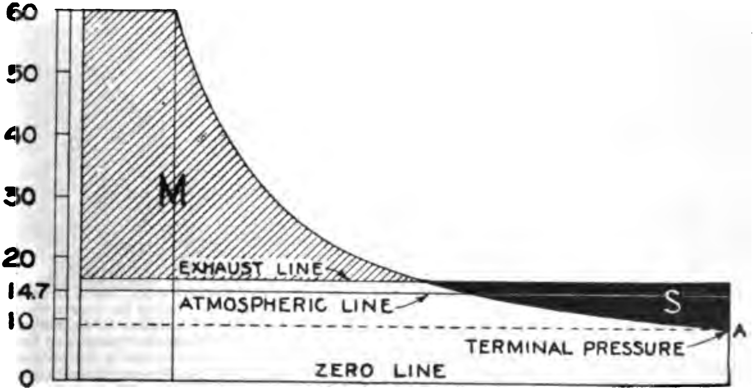


FIG. 79.—Diagram illustrating effect of expanding to a terminal pressure less than the exhaust pressure. In the above card, representing non-condensing operation, steam is expanded to A, below the exhaust line, giving the negative area S, which must be subtracted from M, to obtain the effective work area.

Expansion Constant.—To determine the pressure at any point of the stroke, use is made of a constant found by multiplying the volume of steam at cut off by the initial pressure.

For instance, if steam at 80 lbs. absolute pressure be cut off when the piston has moved 10 inches of the stroke, then

$$\text{volume} \times \text{pressure} = \text{constant}$$

substituting the above values

$$10 \times 80 = 800$$

Rule. *The pressure at any point of the stroke equals the expansion constant divided by the volume at that point.*

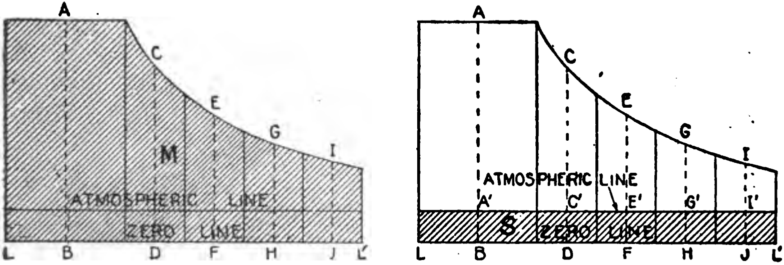
Thus, when the piston has passed through 20 inches of the stroke the pressure at that point is

$$800 \div 20 = 40 \text{ lbs. absolute.}$$

Rule. *The volume corresponding to any pressure is equal to the expansion constant divided by the pressure at that point.*

Thus when the pressure has decreased to 40 lbs. absolute, the volume corresponding as measured by the piston movement is

$$800 \div 40 = 20 \text{ inches}$$



Figs. 80 and 81.—Theoretical cards illustrating ***mean forward pressure*** and ***back pressure.*** The two cards are the same as the card in fig. 78. If in fig. 78 an ordinate be drawn through the middle of each of the areas C D C′ D′, D′ H F D, etc., they will appear in fig. 80 as the dotted vertical lines A B, C D, E F, etc. ***The mean forward pressure*** represented by the area of M ÷ its length L L′ × the pressure scale, is equal to the average of the ordinates, that is, their sum divided by the number, and multiplied by the pressure scale, or $\frac{(AB + CD + EF + GH + IJ)}{5} \times$ pressure scale. Similarly in fig. 81, ***the back pressure,*** = area S ÷ its length L L′ × pressure scale $= \frac{A'B + C'D + E'F + G'H + I'J}{5} \times$ pressure scale, or since in this case all are of the same length, back pressure = height of S × pressure scale.

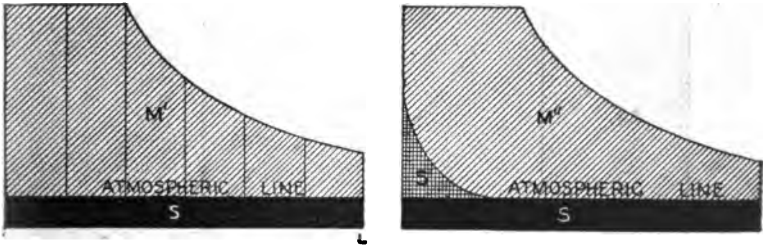
Mean Effective Pressure.—In the diagrams, figs. 82 and 83, the effective pressure which tends to move the piston is clearly *the difference between the* **steam pressure** *acting on one face of the piston and the* **atmospheric pressure** *acting on the other face in the opposite direction.*

The steam pressure acting in the direction in which the piston moves is called the *forward pressure,* and any pressure as that of the atmosphere acting on the opposite face, and opposing the movement of the piston is called the *back pressure.*

If steam be expanded as in the diagram, its pressure will vary, hence it is necessary to find an average or *mean pressure* which shall be the equivalent of this varying forward pressure.

The *mean effective pressure* is equal to the difference between the *mean forward pressure* and the *mean back pressure.* That is, the mean effective pressure, or

*M.E.P.=mean forward pressure—mean back pressure.**



FIGS. 82 and 83.—Theoretical cards illustrating ***mean effective pressure:*** fig. 82, constant back pressure; fig. 83, variable back pressure. Since the back pressure directly opposes the forward pressure, evidently the net or actual pressure tending to move a piston, or mean effective pressure is the difference between these two pressures, that is, *M. E. P.=mean forward pressure — back pressure.* In fig. 80 the mean forward pressure is figured from the area M, and in fig. 81, the back pressure from the area S, hence, the mean effective pressure must depend on the difference of these two areas, that is M — S, or M' as shown in fig. 82, *when the back pressure is constant.* Where compression is taken into account, as in fig. 83, evidently *M. E. P.=mean forward pressure — mean back pressure,* but in the figure, the mean back pressure is figured from area S+area S', hence, in this case M. E. P. depends on the difference between area M in fig. 80 and areas S+S' in fig. 83, and giving the area M'' where *average ordinate×pressure scale=M. E. P.*

If the distance of all points on the expansion line from the vacuum line be measured, and the sum of these distances be divided by their number, the quotient will equal the mean forward pressure; from which is deducted the back pressure, *both in pounds absolute,* and the result will be the mean effective pressure.

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*NOTE.—In the theoretical card, fig. 82, the back pressure S, is constant, but in practice it varies, hence in the actual card, the mean effective pressure =mean forward pressure—*mean* back pressure. It should be noted that in a theoretical card, taking into account compression, the *mean* back pressure must be subtracted from the mean forward pressure to obtain the M. E. P.

Hyperbolic Logarithms.—In the diagram, fig. 84, the *exact value* of the area S, may be readily obtained by referring to a table of hyperbolic logarithms; for, since the curve of expansion is an hyperbola, the hyperbolic logarithm of the number of expansions expresses the relation between the area S, during expansion and the area M during admission. That is, if

admission area M = *unity*

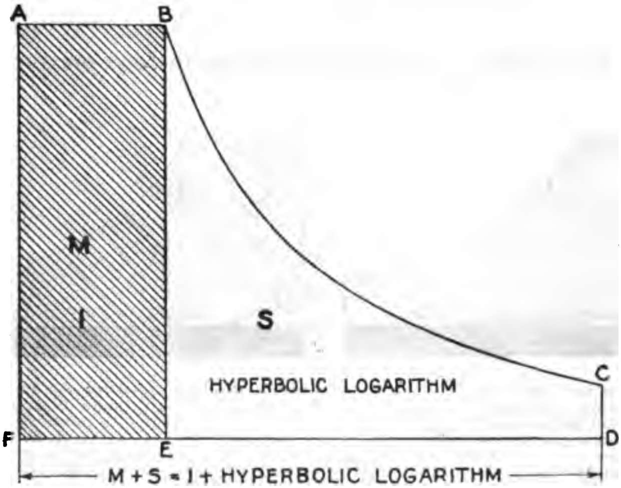


FIG. 84.—Reproduction in part of fig. 71, showing the application of the hyperbolic logarithm in finding the mean effective pressure.

expansion area S = *hyperbolic logarithm*

and

total area M + S = 1 + *hyperbolic logarithm*

Thus, if steam be cut off at one-quarter stroke, it is expanded to 4 times its original volume; then if the area during admission = 1, the area during expansion = the hyperbolic logarithm of 4.

Now, turning to the table of hyperbolic logarithms on page 71, the hyp. log. of 4 is 1.3863. This is the theoretical gain by expansion, that is, if 1 represent the work M, done before expansion, the work S, done during expansion is 1.3863 times greater than the work M, done before expansion.

The total work done during the stroke is equal to

$$M+S=1+\textit{hyp. log.}\ 4=1+1.3863=2.3863$$

Hence, over twice the work is done by admitting steam one-quarter

Table of Hyperbolic Logarithms

No.	Hyp. log.	No.	Hyp. log.	No.	Hyp. log.	No.	Hyp. log.
1.1	0.0953	4.5	1.5041	7.9	2.0669	19.0	2.9444
1.2	0.1823	4.6	1.5261	8.0	2.0794	20.0	2.9957
1.3	0.2624	4.7	1.5476	8.1	2.0919	21.0	3.0445
1.4	0.3365	4.8	1.5686	8.2	2.1041	22.0	3.0910
1.5	0.4055	4.9	1.5892	8.3	2.1163	23.0	3.1355
1.6	0.4700	5.0	1.6094	8.4	2.1282	24.0	3.1781
1.7	0.5306	5.1	1.6292	8.5	2.1401	25.0	3.2189
1.8	0.5878	5.2	1.6487	8.6	2.1518	26.0	3.2581
1.9	0.6419	5.3	1.6677	8.7	2.1633	27.0	3.2958
2.0	0.6931	5.4	1.6864	8.8	2.1748	28.0	3.3322
2.1	0.7419	5.5	1.7047	8.9	2.1861	29.0	3.3673
2.2	0.7885	5.6	1.7228	9.0	2.1972	30.0	3.4012
2.3	0.8329	5.7	1.7405	9.1	2.2083	31.0	3.4340
2.4	0.8755	5.8	1.7579	9.2	2.2192	32.0	3.4657
2.5	0.9163	5.9	1.7750	9.3	2.2300	33.0	3.4965
2.6	0.9555	6.0	1.7918	9.4	2.2407	34.0	3.5263
2.7	0.9933	6.1	1.8083	9.5	2.2513	35.0	3.5553
2.8	1.0296	6.2	1.8245	9.6	2.2618	36.0	3.5835
2.9	1.0647	6.3	1.8405	9.7	2.2721	37.0	3.6109
3.0	1.0986	6.4	1.8563	9.8	2.2824	38.0	3.6376
3.1	1.1312	6.5	1.8718	9.9	2.2925	39.0	3.6636
3.2	1.1632	6.6	1.8871	10.0	2.3026	40.0	3.6889
3.3	1.1939	6.7	1.9021	10.5	2.3513	41.0	3.7136
3.4	1.2238	6.8	1.9169	11.0	2.3979	42.0	3.7377
3.5	1.2528	6.9	1.9315	11.5	2.4430	43.0	3.7612
3.6	1.2809	7.0	1.9459	12.0	2.4849	44.0	3.7842
3.7	1.3083	7.1	1.9601	12.5	2.5262	45.0	3.8067
3.8	1.3350	7.2	1.9741	13.0	2.5649	46.0	3.8286
3.9	1.3610	7.3	1.9879	13.5	2.6027	47.0	3.8501
4.0	1.3863	7.4	2.0015	14.0	2.6391	48.0	3.8712
4.1	1.4110	7.5	2.0149	15.0	2.7081	49.0	3.8918
4.2	1.4351	7.6	2.0281	16.0	2.7726	50.0	3.9120
4.3	1.4586	7.7	2.0412	17.0	2.8332		
4.4	1.4816	7.8	2.0541	18.0	2.8904		

NOTE.—Hyperbolic or Naperian logarithms are common logarithms multiplied by 2.3025851.

stroke and expanding four times than by admitting steam one-quarter stroke and exhausting at that point into the atmosphere.

If the distance F E be called 1, then 4 expansions F D = 4. Now, if the total area M + S, be divided by F D, or 4, it will give the height of a rectangle whose area = M + S, and the height of this rectangle will represent the mean effective pressure, hence:

Rule.—*To find the mean effective pressure, multiply the initial pressure* **in lbs. absolute** *by 1 + hyp. log. of the number of expansions, and divide by the number of expansions. From the quotient, subtract the* **absolute** *back pressure.*

In the form of an equation the mean effective pressure or

$$\text{M.E.P.} = \frac{\textit{initial pressure abs.} \times 1 + \textit{hyp.log.no. of expansions}}{\textit{number of expansions}} - \textit{back pressure abs.}$$

or, expressed in the usual symbols

$$\text{M. E. P.} = \frac{\text{P} \times 1 + \text{hyp. log. } r}{r} - \text{B. P.}$$

It should be remembered that the initial pressure P, and back pressure B. P., are taken *in lbs.* **absolute.**

Example.—What is the mean effective pressure, with 80 lbs. initial gauge pressure, one-third cut off, 16 lbs. absolute back pressure?

Initial pressure absolute = 80 + 14.7 = 94.7 lbs.

Number of expansions $= 1 \div {}^1\!/_3 = 1 \times {}^3\!/_1 = 3$.

Hyp. log. of 3 (from table page 71) = 1.0986.

1 + hyp. log 3 = 1 + 1.0986 = 2.0986.

$$\text{Mean effective pressure} = \frac{94.7 \times 2.0986}{3} - 16 = 50.2 \text{ lbs.}$$

Ques. In the operation of an engine is there as much advantage from working steam expansively as the above calculations indicate?

Ans. **No;** *it is not possible in steam engines to convert all the energy of the steam into useful work. There are various losses due to leakage, radiation, condensation and other causes, all of which tend to make the actual mean effective pressure obtained in an engine less than that calculated*, as shown in fig. 85.

Table for Finding Mean Pressure

Number of expansions	$\frac{1+\text{hyp. log. } r}{r}$	Number of expansions	$\frac{1+\text{hyp. log. } r}{r}$
1.0	1.00	11.0	0.309
1.5	0.937	12.0	0.290
2.0	0.847	13.0	0.274
2.5	0.766	14.0	0.260
3.0	0.700	15.0	0.247
3.5	0.644	16.0	0.236
4.0	0.597	17.0	0.226
4.5	0.556	18.0	0.216
5.0	0.522	19.0	0.208
5.5	0.492	20.0	0.200
6.0	0.465	21.0	0.192
7.0	0.421	22.0	0.186
8.0	0.385	23.0	0.180
9.0	0.355	24.0	0.174
10.0	0.330	25.0	0.169

How to Use the Table.—The mean pressure is obtained for any number of expansions by multiplying the initial pressure *absolute* by the factor given.

Example.—What is the mean pressure of steam for 100 lbs. initial gauge pressure, and one-quarter cut off?

100 lbs. gauge pressure $=100+14.7=114.7$ lbs. absolute pressure; $\frac{1}{4}$cut-off $=1\div\frac{1}{4}=4$ expansions.

In the table the factor for four expansions is .597, from which the mean pressure is

$$114.7\times.597=68.5 \text{ lbs.}$$

To find the *mean effective pressure*, the *absolute* back pressure is subtracted from the mean forward pressure just obtained.

Thus in the example just given, if the engine be running non-condensing, and exhausting against a back pressure of 2 lbs. gauge, then the absolute back pressure $=2+14.7=16.7$ and the mean effective pressure $=68.5-16.7=51.8$ lbs.

Again, if the engine be running condensing with a 28 inch vacuum, the absolute pressure corresponding to this vacuum is, from the steam table on page 40, .946 lbs. absolute, and the mean effective pressure is

$$68.5-.946=67.55 \text{ lbs.}$$

Diagram Factor.—From the answer to the last question, (page 73), it is seen that no such results are obtained in the actual engine, as in theoretical calculations. The diagram, fig. 84 is known as a theoretical (indicator) card and represents a perfect performance, assuming hyperbolic expansion, that is, if the valves

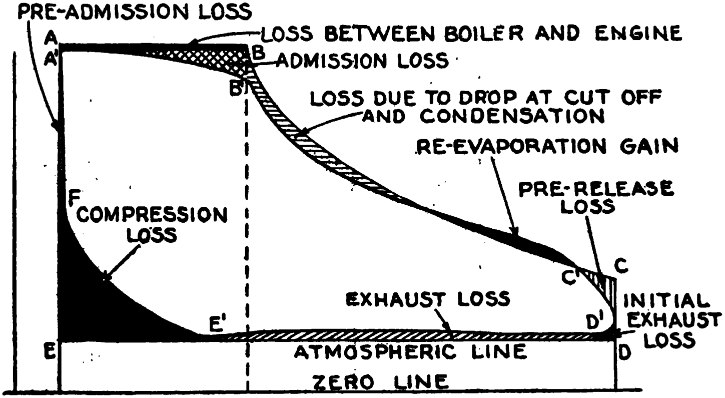
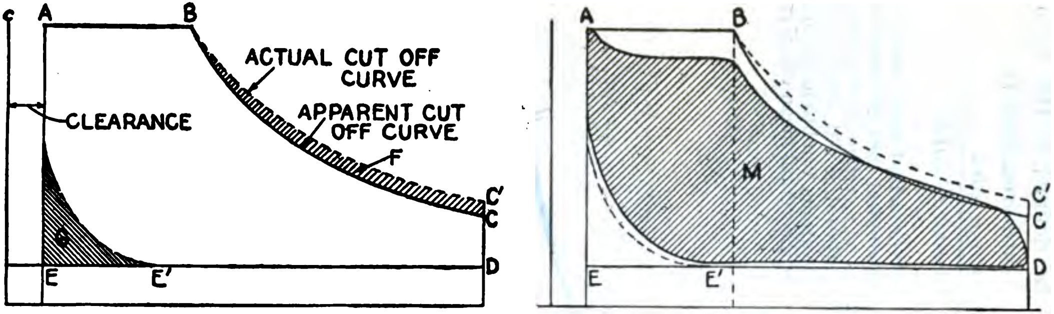


FIG. 85.—Comparison of theoretical and actual cards showing the various losses which tend to reduce the area of the actual card, making it in some cases considerably less than that of the theoretical card. In the figure, A B C D E A, is the theoretical card and A′ B′ C′ D′ E′ F A′, the actual card. ***In practice,*** an initial loss occurs in getting the steam from the boiler to the engine making the beginning of the actual card at A′ instead of A. ***During admission,*** the pressure drops because of friction through ports and passages, becoming very pronounced at cut off by "wire drawing." ***During expansion,*** the curve at first is below the theoretical because of loss of pressure at cut off, later condensation, and re-evaporation causes it to rise slightly above before pre-release. ***During pre-release*** the pressure drops very quickly, but does not reach exhaust pressure until the piston has begun the exhaust stroke. ***During exhaust,*** the pressure is always greater in the cylinder than the external back pressure of the atmosphere or condenser. ***During compression,*** sometimes considerable area of card is lost. ***During pre-admission*** the steam is retarded in rising to admission pressure because of wire drawing, this loss during this period is very small, and in some cases not noticeable especially if there be liberal lead because the piston is practically stationary.

could open and close *instantly*, avoiding "wire drawing" or loss of pressure while not fully open or closed.

If there were no condensation, or any other condition causing a loss of pressure, the diagram, fig. 84, would represent the performance of an engine working under such conditions.

In practice, as before stated *no such results are possible*, a diagram of less area (which means less work) being obtained. This diagram is obtained by means of an indicator and is called the actual or indicator diagram to distinguish it from the theoretical diagram which is constructed from the calculated performance. The relative value of these diagrams is expressed by a coefficient



FIGS. 86 and 87.—Diagrams illustrating **the diagram factor.** In fig. 86, the theoretical diagram, A B C D E does **not consider** clearance and compression; it may be for certain purposes *modified* to allow for these items. Thus, in the figure, not considering clearance, the admission would be represented by A B, giving the expansion curve B C. Now taking clearance and compression into account, if *c* A, represent the clearance volume, then admission would be represented by *c* B, instead of A B, giving the expansion curve B C′ *above* B C, thus giving to the card the additional area represented by the shaded portion F, but the effect of introducing compression would be to take away from the card the shaded area G, giving the modified theoretical card A B C′ D E′ A. ***In many engines*** the effect of clearance and compression is to make these areas about equal, thus, generally it becomes unnecessary to modify the theoretical card. In fig. 87, the shaded diagram M, is an actual diagram compared with the theoretical, and modified theoretical diagrams. ***By measurement*** the area of the actual diagram M, is 1.47 sq. ins.; the theoretical card A B C D E A, = 1.8 sq. ins., and the *modified* theoretical card A B C′ D E′ A, = 1.84 sq. ins. From which, the diagram factor referred to theoretical card = 1.47 ÷ 1.8 = .82, and referred to the modified theoretical card = 1.47 ÷ 1.84 = .799.

called the diagram factor, which may be defined as the ratio of the actual card area to the theoretical card area, that is

$$\textit{diagram factor} = \frac{\textit{area of actual card}}{\textit{area of theoretical card}}$$

Remembering that the work represented by the actual card, that is, its area, is always *less* than the area of the theoretical card, it must be evident that if an engine cylinder be proportioned for a certain horse power, based on mean effective pressure of the theoretical card, it will, when built and tested, develop *less* power because of the various conditions before mentioned which tend to reduce this pressure, or theoretical mean effective pressure as it is called. Accordingly, since the power developed by an engine at any given speed is directly proportional to the mean effective pressure actually obtained in operation, the designer, after constructing a theoretical card for the working conditions of initial pressure, cut off, etc., desired, and finding the theoretical mean effective pressure from this card, multiplies this by the *diagram factor* corresponding to the type of engine being designed. The value thus obtained is called the **expected** ***mean effective pressure***, because the diagram factor, being obtained by comparing the indicator card of a large number of engines of a similar type working under similar conditions with the theoretical card, and therefore representing the allowance which must be made for the various conditions

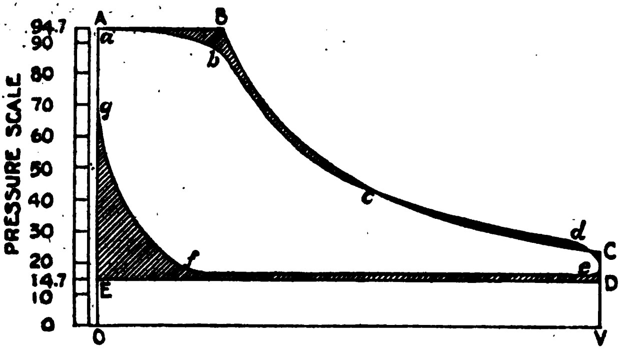


FIG. 88.—Theoretical and "expected" cards of Corliss engine operating under conditions given in example on page 97. After drawing the theoretical card A B C D E, to correspond with the given operating condition, the designer inscribes or sketches within the expected card making such allowance for the various losses as his experience and judgment dictates. The M. E. P. can be obtained, 1, by finding the diagram factor and multiplying it by the theoretical M. E. P., or 2, by finding the expected M. E. P. direct from the expected card. Clearly, the first method would be a waste of time, unless, the designer desire to check the accuracy of his judgment by comparing the diagram factor, with diagram factors of other similar engines already built and operating under similar conditions.

tending to reduce the theoretical mean effective pressure, gives, as near as can be calculated, the actual or "expected" mean effective pressure, when multiplied by the theoretical mean effective pressure.

Example.—The theoretical mean effective pressure of a given theoretical card is 40 lbs. What mean effective pressure must be used in designing an engine to develop a given horse power from the theoretical card if the diagram factor be .85? The mean effective pressure to be used in obtaining the cylinder dimensions, or

$$\text{expected M. E. P.} = 40 \times .85 = 34 \text{ lbs.}$$

It should be noted that if the diagram factor were disregarded and 40 lbs. taken as the M. E. P., then the actual engine, if calculated for, say, 100 horse power at 300 revolutions per minute, would develop approximately only

$$100 \times \frac{34}{40} = 85 \text{ horse power}$$

being $100 - 85 = 15$ horse power short of the calculated power.

If the theoretical mean pressure be calculated, and the necessary corrections made for *clearance* and *compression*, according to Seaton the expected mean effective pressure may be found by multiplying the results by the factor in the first column of the following table:

Diagram Factors

Particulars of Engine	Diagram Factor	
Expansive engine, special valve gear, or with a separate cut off valve, cylinders jacketed........	.94	.9
Expansive engine having large ports, etc., and good ordinary valves, cylinders jacketed................	.9 to .92	.86 to .88
Expansive engines with the ordinary valves and gear as in general practice and unjacketed............	.8 to .85	.77 to .82
Compound engines, with expansion valve to H. P. cylinder; cylinders jacketed, and with large ports, etc....................................	.9 to .92	.86 to .88
Compound engines, with ordinary slide valves, cylinders jacketed, and good ports, etc..	.8 to .85	.77 to .82
Compound engines as in general practice in the merchant service, with early cut off in both cylinders without jackets and expansion valves............	.7 to .8	.67 to .77
Triple expansion engines, with ordinary slide valves, good ports, unjacketed, moderate piston speed....	.65 to .7	.62 to .67
Fast running engines of the type and design usually fitted in war ships..........................	.6 to .7	.58 to .67

If no correction be made for the effects of clearance and compression, and the engine is in accordance with general modern practice, the clearance and compression being proportionate, then the theoretical *effective* mean pressure may be found by multiplying the results in the last column by .96, giving the values in the second column.

Horse Power.—This unit, as before stated was introduced by James Watt to measure the power of his steam engines and which he considered as being the power of a strong London draught horse to do work *for a short time*. This he estimated to be equal to 33,000 foot pounds per minute.* One horse power then, or

one H. P. = 33,000 *ft. lbs. per minute*

which is the accepted standard.

According to definitions, and the manner in which it is determined, horse power may be classed as

1. Nominal (N. H. P.);
2. Theoretical (T. H. P.);
3. Indicated (I. H. P.);
4. Brake (B. H. P.);
5. S. A. E.;
6. Electrical (E. H. P.); etc.

Nominal Horse Power.—In the early days Watt, according to Seaton, found that the mean pressure usually obtained in the cylinders of his engines was 7 lbs. per sq. ins. He had also found the proper piston speed at $128 \times \sqrt[3]{\text{stroke}}$ per minute, and his engines were arranged to work at this speed, so that he estimated the power which would be developed when at work to be

$$\dagger \text{N. H. P.} = A \times 7 \times 128 \times \sqrt[3]{S}$$

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*NOTE.—James Watt was early asked by would be purchasers as to how many horses his engines would replace. To obtain data as to actual performance in *continuous* work, he experimented with powerful brewery horses, and found that one traveling at 2½ miles per hour, or 220 feet per minute, and harnessed to a rope leading over a pulley and down a vertical shaft, could haul up a weight averaging 100 lbs., equaling 22,000 foot pounds per minute. To give good measure, Watt increased the measurement by 50 per cent., thus getting the familiar unit of 33,000 *foot pounds* per minute.

†NOTE.—The power calculated by the formula above was called "nominal," because the engine was described as of that power, and in practice that power was actually obtained. However, when the boiler could be constructed so as to supply steam above atmospheric pressure, and the engine was run with more strokes per minute than before, the power developed exceeded the nominal power, thus causing the nominal horse power rating to be discontinued.

in which A = area of piston in sq. ins.; S = number of strokes per minute. The term nominal horse power is now obsolete and is only of historical interest.

Indicated horse power.—This is *the actual power developed by an engine as calculated from the indicator card.* It should be understood that it represents the power developed at the instant the card was taken, and not necessarily at any other instant*. It should be carefully noted that the indicated horse power of an engine does not represent the power delivered, being in excess of the power delivered by an amount equal to the power lost by friction in the engine.

Brake Horse Power.—By definition, *the actual power delivered by an engine as determined by a brake test.* This is sometimes called the *delivered horse power*, and is always *less* than the indicated horse power by an amount equal to the power absorbed by friction in the engine.†

S. A. E. Horse Power.—In order to reduce all automobile engines to a common basis of rating for determining the class of license required, the commissioners of motor vehicles have adopted a rule known as the S. A. E. horse power formula, and which assumes that all gas engines will deliver or should deliver their rated power at a piston speed of 1,000 feet per minute, mean effective pressure of 90 lbs. per sq. ins., and mechanical efficiency of 75 per cent. It should be understood that a formula based on such data is worthless for obtaining the actual horse power of an engine and should only be used for the purpose for which it is intended.

Electrical Horse Power.—It is desirable to establish the relation between *watts* and *foot pounds* in order to determine the *capacity* of an electric generator or motor in terms of *horse power*.

One watt is equivalent to one joule per second or 60 joules per minute. One joule in turn, is equivalent to .7374 ft. lbs., hence 60 joules equal:

$$60 \times .7374 = 44.244 \text{ ft. lbs.}$$

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*NOTE.—It should be understood that in operation the power developed is continually varying, and, very strictly speaking, may be said never to be the same during any appreciable interval of time.

†NOTE.—The ratio between the indicated and brake horse power of an engine, that is brake horse power ÷ indicated horse power represents the mechanical efficiency of the engine; this should not be confused with the thermal efficiency, or heat units converted into useful work ÷ heat units supplied to the engine.

Since one horse power = 33,000 ft. lbs. per minute, the electrical equivalent of one horse power is

$$33{,}000 \div 44.244 = 746 \text{ watts.}$$

or,

$$\frac{746}{1{,}000} = .746 \text{ kilowatts (K. W.)}$$

Again, one kilowatt or 1,000 watts is equivalent to

$$1{,}000 \div 746 = 1.34 \text{ horse power.}$$

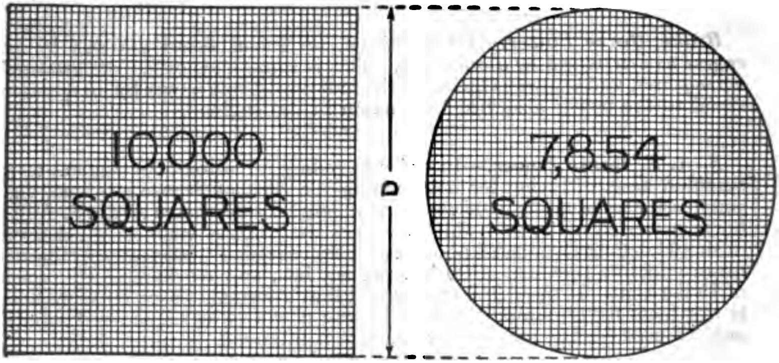

FIG. 89.—Diagram illustrating why the decimal .7854 is used to find the area of a circle. If the square be divided into 10,000 parts or small squares, a circle having a diameter D, equal to a side of the large square will contain 7854 small squares, hence, if the area of the large square be 1 sq. in., then the area of the circle will be 7854+10,000 or .7854 sq. ins., that is, area of the circle $= .7854 \times D^2 = .7854 \times D \times D = .7854 \times 1 \times 1 = .7854$ sq. ins.

How to Calculate Horse Power.—There are various formulæ for calculating the power of engines, and the student should endeavor to understand the principles upon which they are based rather than simply committing them to memory. Before taking up these formulæ a few preliminary considerations are necessary.

Ques. Why is the decimal .7854 used to ascertain the area of a circle or piston?

Ans. Because it represents the relation between a circle and circumscribed square.

This relation is clearly shown in fig. 89.

Ques. What is understood by the term piston speed?

Ans. It is the total distance traveled by the piston of an engine in one minute—not the actual velocity at any given instant of time.

Ques. How is the piston speed obtained?

Ans. RULE: Multiply twice the number of revolutions per minute by the stroke of the engine in inches and divide the product by 12 to reduce to feet.

Thus, an engine having a stroke of 6 inches and running 500 revolutions per minute is said to have a piston speed of

$$\frac{2\times 6\times 500}{12}=500 \text{ feet per minute.}$$

Ques. What is the usual method of calculating the horse power of an engine?

Ans. RULE: *Multiply the mean effective pressure in lbs. per square inch by the area of piston in square inches and multiply the product by the length of stroke* **in feet,** *and by the number of strokes per minute* (*twice the number of revolutions*); *divide this last product by* 33,000 *and the answer will be the horse power for a* **double acting** *engine.*

This method which is very generally used is expressed as a formula as follows:

$$\text{H. P.}=\frac{2\times P\times L\times A\times N}{33{,}000}=\frac{2(.7854D^2)PLN}{33{,}000} \quad . \; . \; . \; . \; . \; (1)$$

NOTE.—*Horse power expressed in thermal units.*—Since 1 B. t. u. is equivalent to 777.52 ft. lbs. (Marks and Davis), and one horse power = 33,000 ft. lbs. per minute, then *one horse power* = 33,000 ÷ 777.52 = 42.44 B. t. u. *per minute.*

in which

P = mean effective pressure in lbs. per sq. ins.;
L = length of stroke in *feet;*
A = area of piston in sq. ins. = .7854 × diameter of piston squared;
N = number of *revolutions* per minute;
D = diameter of piston.

It should be noted that the numerator represents the total ft. lbs. done by the engine in one minute; the figure 2 is introduced because in the double acting engine there are two *power* strokes each revolution. The denominator or 33,000 is the foot pounds per one minute for one horse power.

Example.—What is the horse power of a 5×6 engine running at 500 revolutions per minute and 50 lbs. mean effective pressure?

Substituting these values in the formula, and remembering that the area A of the piston = .7854 × its diameter squared,

$$\text{H. P.} = \frac{2\times(.7854\times5^2)\times50\times\frac{6}{12}\times500}{33{,}000} = 14.87$$

Ques. What is the objection to the formula just given?

Ans. It involves a considerable waste of time in making the calculation.

Since the stroke of an engine is usually given in inches instead of feet, and the revolutions per minute instead of the piston speed, the formula just given evidently involves extra calculations for these items as well as the extra multiplication and division introduced because of the constants. Its use therefore is about as laborious as multiplying and dividing fractions without reducing them to their lowest terms.

The author strongly recommends that the formula just given be not used in the form given but reduced to its lowest terms as follows:

$$\text{H. P.} = \frac{2\,P\,L\,A\,N}{33{,}000} = \frac{2\times P\times\frac{L}{12}\times.7854\times D^2\times N}{33{,}000} = \frac{.1309\times PLD^2N}{33{,}000}$$
$$= .000003967\ P\,L\,D^2N$$

Using the constant .000004 instead of .000003966 which is near enough for ordinary calculations, and changing the order of the factors, the formula becomes

$$\text{H. P.} = .000004\ D^2\,L\,N\,P \quad . \quad . \quad . \quad (2)$$

Example.—What is the horse power of the engine in the previous example (running under the same conditions), as calculated by formula (2)? Substituting the given value in (2)

$$H. P. = .000004 \times 5^2 \times 6 \times 500 \times 50 = 15$$

Comparing the two formulæ, 15 h. p. is here obtained instead of 14.87, the error introduced by using the constant .000004 instead of .000003966, being only

$$15 - 14.87 = .13 \text{ horse power or } \frac{86}{100} \text{ of } 1\%$$

This short formula (2) is very valuable to those who have frequent occasions to calculate horse power. The power of any engine on a basis of of 500 revolutions and 50 lbs. mean effective pressure can be very quickly

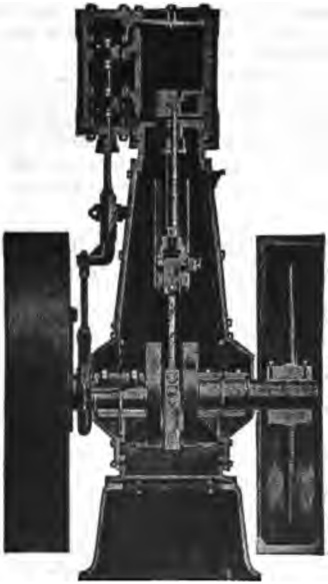

FIG. 90.—View of Buffalo small vertical piston valve stationary engine. The main bearings are ring oiling and receive their supply of lubrication from the bed by the rotation of the crank disk; they are carried on a heavy plate bolted to the frame. Removable heads surmounting them shut out all the dust and grit and allow of ready access for adjustment. A false head forms a chamber to prevent condensation getting into the engine bed. This is a so called "square" engine, that is the stroke is the same length as the piston diameter. The 5 × 5 size at 475 r. p. m. is rated at 12 horse power, and the 6 × 6 at 450 r. p. m., 18 horse power. Compare these ratings with the 5 × 6 engine given in the example on page 82.

found with this formula and the method of using it, as given below, will firmly fix it in mind, though as before stated, the author does not recommend memorizing formulæ but instead, the acquirement of a knowledge of principles upon which they depend.

Now, for a quick calculation of the horse power of the engine in the previous example,

1. Write down the cylinder dimensions, squaring the diameter

$$(5^2 \times 6)$$

2. Disregard the decimal point and write 4 instead of .000004

$$4 \times (5^2 \times 6)$$

3. Insert the revolutions per minute and the mean effective pressure

$$4 \times (5^2 \times 6) \times 500 \times 50$$

The product of these factors is the horse power when the decimal point is inserted in the right place.

4. Since the product of the first and last two factors is 100,000, disregarding the ciphers, only the factors inside the parenthesis need be considered to obtain the horse power, thus

$$5^2 \times 6 = 5 \times 5 \times 6 = 150 \quad \ldots \ldots \ldots \ldots \quad (3)$$

It remains only to insert the decimal point, which is determined from the sense of proportion, that is, any one familiar with engines would know that a 5×6 engine running at 500 R. P. M. and 50 lbs. mean effective pressure does not develop 150 h. p. as written in equation (3); neither does it develop only 1.5 h. p.; it must then develop 15 h. p.

From the foregoing it must be evident that to obtain the horse power of any engine running at 500 revolutions per minute and 50 lbs. mean effective pressure it is only necessary to consider the dimensions of the cylinder and to point off one place, or multiply by .1 as expressed in the following formula,

$$\textbf{H. P.} = .1 \times \textbf{diameter piston}^2 \times \textbf{stroke} \quad . \ . \quad (4)$$

diameter and stroke being taken in inches. Expressed as a rule the formula becomes:

Rule.—*For* 500 *revolutions per minute, and* 50 *lbs. per sq. in. M. E. P., square the diameter and multiply by the stroke, both in inches; multiply the product by* .1, that is, point off one place.

Ques. How is the horse power obtained by (4) for other than 500 R. P. M. and 50 lbs. M. E. P.?

Ans. By multiplying the result obtained in (4) by the ratio between the given R. P. M. and 500, and the given M. E. P. and 50.

Expressed as a formula (4) becomes

$$\text{H. P.} = (\text{h.p. at 500 r. p. m. and 50 lbs. m. e. p.}) \times \frac{\text{R. P. M.}}{500} \times \frac{\text{M. E. P.}}{50}. \quad (5)$$

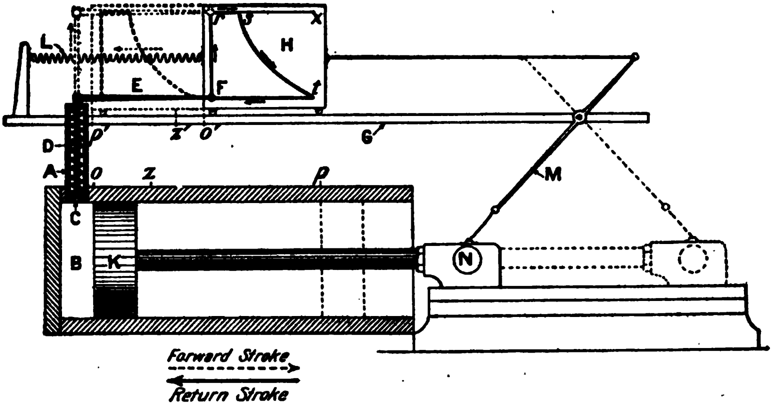

FIG. 91.—Working principles of the indicator. In the figure, A, is a small cylinder screwed into the engine cylinder and opening into the clearance space B. C, is a piston working within A, against the pressure of the steam in B, by means of the tension of the spring D. E, is a horizontal arm attached to the rod of the piston C, and carrying on its outer end a pencil point F. G, is a carrier bar upon which a board H, carrying a sheet of paper is moved back and forth in a direction opposite to that of the piston of the engine, by means of the spring L, and the lever M, the upper end of the latter being attached by a cord to the movable board and the lower end to some part of the piston rod such as the crosshead N. ***In operation***, assume the piston K, to be at its inner dead center *o*, and the clearance space B, to be empty. The piston C, will be down, and pencil point at F. Now, if steam be admitted to B, the increasing pressure will drive the piston C, upward, carrying the pencil vertically from F, to *r*, until the pressure in the clearance space is sufficient to move the piston. If this pressure be kept constant while the piston travels from *o*, to *p*, and moves the board H, through a corresponding distance from *o'*, to *p'*, the pencil will trace the line *r x*. But ordinarily, the pressure is not kept constant, the supply of steam being stopped when the piston has traveled some part of its forward stroke. Assume that the supply of steam be stopped when the piston has traveled a distance equal to one-quarter the length of its full stroke, or to *z*. The movement of the piston from *o*, to *z* will carry the board from *o'*, to *z'*, and as the pressure is kept constant up to this point, the pencil will trace a horizontal line from *r*, to *s*, the *cut off point*. The continued advance of the piston will move the board towards *p'*, and as it will also increase the volume of the steam, the pressure in the engine cylinder will fall, thus relieving the compression on the spring D, and allowing the piston C. to descend. As the result of these operations and movements, the indicator pencil will trace the line *s t*, the point *t* coinciding with the point F, on the diagram when the piston is at its outer dead center *p*, and the board at the limit of its backward movement *p'*. Driven by the stored up energy in the fly wheel, the engine piston will travel from *p*, to *o*, on its return stroke, pulling the

Example.—A 5×6 engine at 500 R. P. M. and 50 lbs. M. E. P. develops 15 horse power. What will be the power at

1. 250 R. P. M. and 50 M. E. P.?
2. 500 R. P. M. and 40 M. E. P.?
3. 400 R. P. M. and 60 M. E. P.?

The factor in the parenthesis of formula (5) being given in the example as 15, the powers developed corresponding to the above running conditions are

1. $\text{H. P.} = 15 \times \frac{250}{500} \times \frac{50}{50} = 7\frac{1}{2}.$

2. $\text{H. P.} = 15 \times \frac{500}{500} \times \frac{40}{50} = \frac{60}{5} = 12.$

3. $\text{H. P.} = 15 \times \frac{400}{500} \times \frac{60}{50} = \frac{3 \times 4 \times 6}{5} = 14.4.$

This method is useful for mental calculation.

Expressed as rules, the two principles upon which the above calculations are based are:

Rule.—*At constant speed, the horse power of an engine varies directly as the mean effective pressure.*

Rule.—*At constant mean effective pressure, the horse power of an engine varies directly as the speed.*

Ques. Give a very short rule for finding the horse power of a single cylinder engine.

Ans. Square the piston diameter and divide by 2.

FIG. 91.—*Continued.*

board from p' to o', and as no pressure exists in the cylinder, the indicator piston will remain down, and the indicator pencil will trace the line t F, and thus complete the ***diagram, the area of which graphically represents*** the ***work done by the engine per rev.*** It should be noted that for simplicity, pre-release and excess back pressure are not considered, steam being assumed to expand to the pressure of exhaust at t. ***In actual indicators,*** the pencil arm E, referred to, instead of being attached in a fixed horizontal position to the upper end of the rod of the indicator piston is replaced by a system of levers which multiplies the motion of the piston, thus permitting the use of indicator cylinders whose pistons have a smaller range of motion. Also, the movable board H, is replaced by a rotatable drum which carries the paper. A spiral spring in the interior of the drum rotates it in a direction opposite to that of the forward stroke of the engine piston, the spring being put into a state of tension, when the drum is rotated in the opposite direction, by means of a cord attached to the engine piston, during the return stroke of the latter. These substitutions allow very compact and efficient mechanical arrangements.

This is correct whenever the product of mean effective pressure and piston speed = 21,000, as in the following combinations:

Mean effective pressure.........	30	35	38.2 (Approx.)	42
Piston speed..................	700	600	550	500

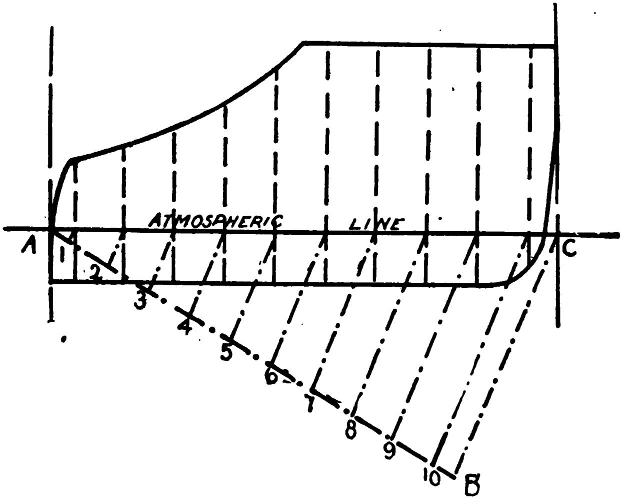

FIG. 92.—Indicator card showing method of finding M. E. P. by **summation of ordinates.** First two lines are drawn perpendicular to the atmospheric line and touching the cards at the ends as shown. On a slanting line starting at the intersection A, of the atmospheric line and the vertical line, a scale is constructed that helps to find the desired ten subdivisions on the length of the diagram, and on which the mean height of each tenth is to be measured. The scale on the slanting line, starting from the point of intersection A, may be made by setting off, first ¼ inch, then, nine ½ inch spaces, and, finally, again ¼ inch, thus making the whole scale five inches long. The end point, B, of this scale is connected by a straight line with the intersection C, of the atmospheric line, and the second vertical line, and lines parallel to B C, are drawn through all the other ten points of the scale, to intersect with the atmospheric line. After drawing vertical lines through all these intersecting points, the ten mean ordinates, or pressures, can be measured, each individually, or in a convenient way, as a sum total, by taking off all the ordinate continuously upon a strip of paper. If this sum total be divided by ten, the mean ordinates for the whole diagram is found. To find the M. E. P., *multiply the mean ordinate by the scale of spring.*

Horse Power Constant.—If it be desired to make a number of horse power calculations of a given engine under different conditions of speed and mean effective pressure to show its power range, it must be evident, that it would be a waste of time to multiply the constants such as piston diameter, stroke, .000004, etc., for each calculation. Accordingly, if all these constants be multiplied, a value is obtained which is called the *horse power constant*, and to obtain the horse power in each case, it is only necessary to multiply the horse power constant by the variable quantity or quantities.

Example.—What is the horse power constant of a 5×6 engine using the formula H. P. = .000004 D^2 L N P. Here .000004; D, diameter = 5, and L, stroke = 6 do not vary, while N and P are variables hence

$$\therefore \text{horse power constant} = .000004 \times 5^2 \times 6 = .0006$$

from which the horse power of this engine at say 500 R. P. M. and 50 lbs. M. E. P. is

$$.0006 \times 500 \times 50 = 15$$

The horse power constant as just found is useful where both N and P, the revolutions and mean effective pressure are considered as "variables." However, in making a "horse power table" for a given engine several sets of calculations are made for a constant value of N or P. That is regarding N as constant, the horse power constant would be

$$.000004\ D^2\ L\ P \quad \ldots\ldots\ldots\ldots \quad (1)$$

or, regarding P as constant, the horse power constant becomes

$$.000004\ D^2\ L\ N \quad \ldots\ldots\ldots\ldots \quad (2)$$

The following examples show the application.

Example.—Calculate the horse power of a 5×6 engine at 30 lbs. mean effective pressure for speeds of 100, 200, 300, 400, and 500 revolutions per minute.

In this case the horse power constant is as expressed in (1) and its value is

$$.000004 \times 5^2 \times 6 \times 30 = .018$$

from which: for 100 r. p. m., h. p. = .018×100 = 1.8; for 200 r. p. m., h. p. = .018×200 = 3.6; for 300 r. p. m., h. p. = .018×300 = 5.4; for 400 r. p. m., h. p. = .018×400 = 7.2; for 500 r. p. m, h. p. = .018×500 = 9.

Example.—Calculate the horse power of the engine in the preceding example at 500 revolutions per minute for mean effective pressures of 40, 50 and 60 lbs. per sq. in.

Here, the horse power constant is as expressed in (2), and its value is

$$.000004 \times 5^2 \times 6 \times 500 = .3$$

from which: for 40 m. e. p., h. p. = .3×40 = 12; for 50 m. e. p., h. p. = .3×50 = 15; for 60 m. e. p., h. p. = .3×60 = 18. The results obtained are tabulated as follows:

Table of Horse Power Constants

For formula H.P. $=.000004D^2LNP$; *constant* $=.000004D^2LP$

Size of cylinder (inches)	*Mean effective pressure*						
	25	30	35	40	50	60	75
1 × 1	.0001	.00012	.00014	.00016	.0002	.00024	.0003
2 × 2½	.001	.0012	.0014	.0016	.002	.0024	.003
3 × 3	.0027	.00324	.00378	.00432	.0054	.00648	.0081
3 × 4	.0036	.00432	.00504	.00576	.0072	.00864	.0108
4 × 4	.0064	.00768	.00896	.01024	.0128	.01536	.0192
5 × 5	.0125	.015	.0175	.02	.025	.03	.0375
5 × 6	.015	.018	.021	.024	.03	.036	.045
6 × 6	.0216	.02592	.0324	.03456	.0432	.05184	.0648
7 × 7	.0343	.04116	.04802	.05488	.0686	.08232	.1029
7 × 9	.0441	.05292	.06174	.07056	.0882	.10584	.1323
8 × 8	.0512	.06144	.07168	.08192	.1024	.16288	.1536
8 × 10	.064	.0768	.0896	.1024	.128	.1536	.192
9 × 9	.0729	.08748	.10206	.13464	.1458	.17496	.2187
9 × 12	.0972	.1164	.13608	.15542	.1844	.2328	.2916
10 × 12	.12	.144	.168	.192	.24	.288	.36
11 × 14	.1694	.20328	.23716	.27104	.3388	.40656	.5082
12 × 12	.1728	.20736	.24182	.27648	.3456	.41472	.5184
13 × 13	.2197	.26364	.30758	.35152	.52728	.92728	.6591
14 × 14	.2744	.32828	.38316	.43804	.5488	.65656	.8232
15 × 15	.3375	.405	.4725	.54	.675	.81	1.0125
16 × 16	.4096	.49052	.57344	.65536	.8192	.98104	1.2288
18 × 18	.5832	.69984	.81648	.93312	1.16640	1.39968	1.7496
20 × 24	.96	1.152	1.344	1.536	1.92	2.304	2.88
22 × 26	1.2584	1.51008	1.66176	2.01344	2.5168	3.02016	3.7752
24 × 30	1.728	2.0736	2.4192	2.7648	3.456	4.1472	5.184
Corliss Sizes							
10 × 24	.24	.288	.336	.384	.48	.576	.72
12 × 24	.3456	.41472	.48384	.55296	.6912	.82944	1.0368
12 × 30	.432	.5184	.6048	.6912	.864	1.0368	1.296
14 × 30	.588	.7056	.8232	.9408	1.176	1.4112	1.764
14 × 36	.7056	.84672	.98784	1.12896	1.4112	1.69344	2.1168
16 × 30	.768	.9216	1.0752	1.2288	1.536	1.8432	2.304
16 × 36	.9216	1.10592	1.29024	1.47456	1.8432	2.21184	2.7648
16 × 42	1.0752	1.29024	1.50528	1.70232	2.1504	2.58048	3.2256
18 × 36	1.1663	1.39936	1.73282	1.86608	3.38261	2.79872	3.49892
18 × 42	1.3608	1.63296	1.90512	2.17728	2.7216	3.26592	4.0818
20 × 36	1.44	1.728	2.016	2.304	2.88	3.456	4.32
20 × 42	1.68	2.016	2.352	2.688	3.36	4.032	5.04
20 × 48	1.92	2.304	2.688	3.072	3.84	4.608	5.76
22 × 42	2.0328	2.43936	2.84592	3.25248	4.0656	4.87872	6.0984
22 × 48	2.3232	2.78784	3.25048	3.71712	4.6464	5.57568	6.9696
22 × 54	2.6136	3.13632	3.65904	4.18176	5.2272	6.27264	7.8408
24 × 42	2.41919	2.90303	3.38687	3.8707	4.83838	5.80606	7.25757
24 × 48	2.7648	3.31776	3.78072	4.42368	5.5296	6.63552	8.2944
24 × 54	3.1104	3.73248	4.35456	4.97664	6.2208	7.46496	9.3312
26 × 48	3.24471	3.89365	4.54259	5.19153	6.48942	7.78731	9.73413
26 × 54	3.6504	4.38048	5.11056	5.84064	7.3008	8.76096	10.9512
26 × 60	4.056	4.8672	5.6784	6.4896	8.112	9.7344	12.168
28 × 48	3.7632	4.50584	5.26848	6.02112	7.5264	9.01168	11.2896
28 × 54	4.2336	5.08032	5.92704	6.77376	8.4672	10.16064	12.7008
28 × 60	4.704	5.6448	6.5856	7.5264	9.408	11.2896	14.112
30 × 48	4.32	5.184	6.048	6.912	8.64	10.368	12.96
30 × 54	4.86	5.832	6.804	7.776	9.72	11.696	14.58
30 × 60	5.4	6.48	7.56	8.64	10.8	12.96	16.2
32 × 48	4.9152	5.89824	6.88128	7.86432	9.8304	11.79648	14.7456
32 × 54	5.5296	6.63552	7.74144	8.84736	11.0592	13.27104	16.5888
32 × 60	6.144	7.33728	8.56026	9.78304	12.288	14.67556	18.432
34 × 54	6.24241	7.49089	8.73937	9.98785	12.48482	14.98178	18.72723
34 × 60	6.935	8.3132	9.7104	11.0976	13.87	16.6264	20.805

Horse Power Table

M. E. P.	Revolutions per minute				
	100	200	300	400	500
30	**1.8**	**3.6**	**4.6**	**7.2**	**9**
40	2.4	4.8	7.2	9.6	**12**
50	3	6	9	12	**15**
60	3.6	7.2	10.8	14.4	**18**

The values in heavy figures are those obtained in the two examples, the other values are obtained by similar calculations, or by a shorter process by applying the rule for variable speed as given on page 86. Thus, having found the value 2.4 h. p. for 100 r. p. m. and 40 lbs. m. e. p., the other values for 40 lbs. m. e. p. would be, applying the rule, 2.4×2= 4.8 h. p. for 200 r. p. m.; 2.4×3=7.2 h. p. for 300 r. p. m., etc.

Ques. For great accuracy, what should be considered in addition to the factors included in the horse power formulæ already given?

Ans. The cross sectional area of the piston rod.

Ques. Why?

Ans. Because it reduces by a small amount the power indicated by the formulæ.

Effect of the Piston Rod on the Power.—It must be evident since the piston rod passes through the stuffing box in the cylinder head, that the area of the piston upon which steam acts at this end is reduced by an amount equal to the cross sectional area of the rod, whereas, on the other side of the piston steam acts on its entire area. Accordingly the power developed at the "crank end" will be less than at the "head end."*

Since this reduction of power is so small and the *range* of power of a steam engine so great, this ordinarily need not be

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*NOTE.—The terms *crank end* and *head end*, mean respectively the end nearest or farthest om the crank or shaft.

considered, and in most cases the extra calculation is a waste of time, however, it is important *that the principle involved be understood.*

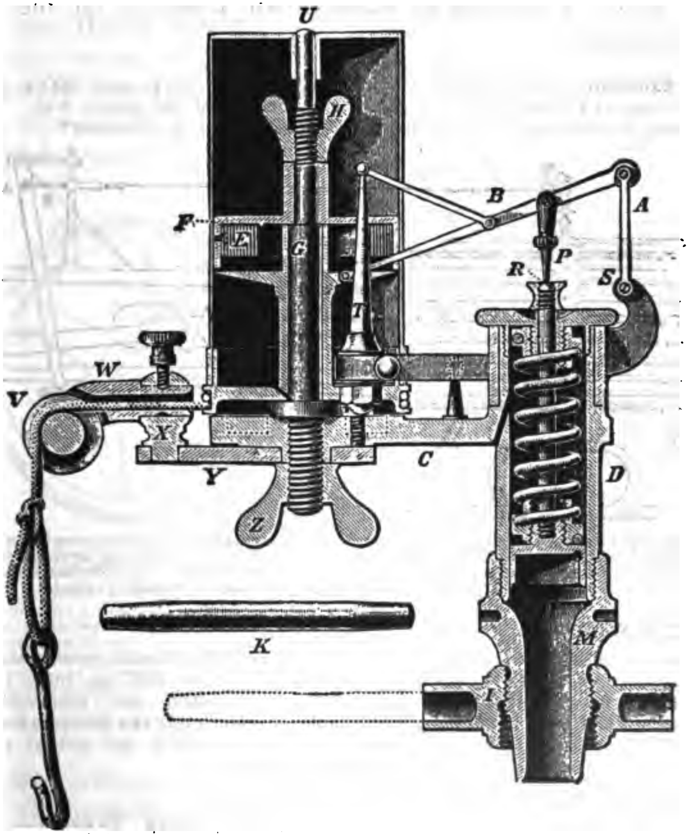


FIG. 93.—Sectional diagram of the indicator. A, is the swinging bar; B, the pencil bar; C, the indicator frame; D, cylinder containing tension spring; E, coiled spring on drum roller; F, revolving cylinder or drum; G, drum pin; H and Z, thumb screws holding drum; I, nut to connect indicator to pipe; K, lever for screwing up I; M, connection between the spring cylinder and pipe; P, piston rod; R, joint; S, pin; T, post for guide of pencil bar; V, guide pulley for cord from reducing lever; W, swivel sleeve for cord; X, swivel pin; Y, support for swivel pin.

To simplify the calculation, the average effect is considered, that is, half the piston rod area is regarded as being removed from each face of the piston, instead of the full area from one face only. The following example will illustrate the methods of calculation.

Example.—A 5×6 engine running at 50 lbs. m. e. p., and 500 r. p. m. develops 15 horse power, neglecting the effect of the piston rod. What power is developed, considering a piston rod 1 inch in diameter?

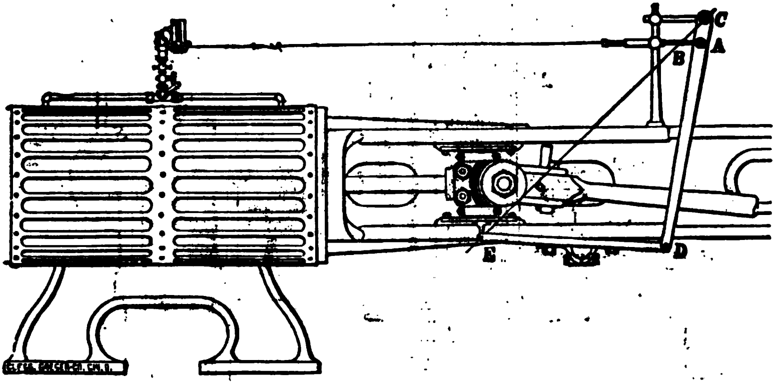


FIG. 94.—One form of reducing lever for an indicator attachment. A, is attachment of the cord; B, end of cord; C, pivot of reducing lever; D, swinging joint of reducing lever; E, point of attachment to cross-head for the link joining reducing lever at D. The indicator is shown at top of cylinder, connected by a three-way cock to pipes from both ends of cylinder.

1st Solution.

From table, or by calculation, area 5″ piston = 19.635; area cross section 1″ piston rod = .7854; ½ piston rod area = .7854 ÷ 2 = .3927 sq. ins. 1% of piston area = 19.635 × .01 = .197, hence ½ piston rod area = .3927 ÷ .197 = .0199, that is 1.99% of piston area. Accordingly the power is reduced 1.99%, or 15 × .0199 = .299 horse power, from which the actual power of the engine is 15 — .299 = 14.7 horse power.

2d Solution.

Area 5″ piston = $\frac{1}{4}\pi D^2 = .7854 \times 5^2 = .7854 \times 5 \times 5 = 19.635$ sq. ins. Similarly, or from table, ½ area of 1″ piston rod = .3927.

Effective piston area = 19.635 — .3927 = 19.2423, say 19.24 sq. ins.

Now, area of piston = $.7854D^2$, from which D^2 = area of piston ÷ .7854, or D = $\sqrt{\text{area of piston} \times .7854} = \sqrt{19.24 \div .7854} = \sqrt{24.497} = 4.95$

Substituting this value of D, and the stroke in formula (4) (page 84) for horse power of an engine running at 50 lbs. m. e. p. and 500 r. p. m.,

$$H. P. = .1 \times 4.95^2 \times 6 = 14.7$$

Brake Horse Power.—This is *the useful horse power delivered by an engine as ascertained by the application of a* **brake** *or* **absorption dynamometer.** The excess of the indicated horse power over that required by the brake, represents the power required to move the engine in overcoming its friction.

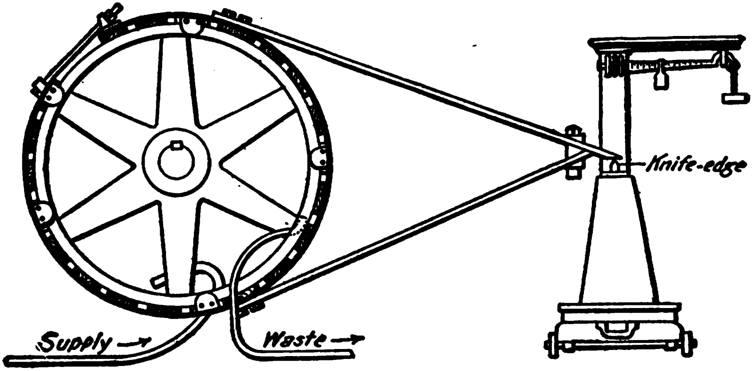


FIG. 95.—Prony brake. It consists of a friction band which may be placed around the fly wheel or the crank shaft, and attached to a lever bearing upon the platform of a weighing scale, as shown. A brake used for testing purposes should be self-adjusting to a certain extent, so as to maintain, automatically, a constant resistance at the rim of the wheel. For comparatively small engines, various forms of rope brake, satisfy this requirement very well. In such cases, a weight is hung to one end of the rope and a spring scale to the other end. The wheel should be provided with interior flanges, holding water for keeping the rim cool. For very high speeds, some form of water friction brake should be employed, as they have the advantage of being self-cooling.

The power of small engines running at very high speeds, is best obtained by a brake test, since indicator cards become disturbed under such conditions, thereby introducing errors. The form of absorption dynamometer generally used for obtaining brake horse power is called the Prony brake (named after its inventor). Its construction is shown in fig. 95.

Formula for Brake Horse Power.—The net work of the engine or horse power delivered at the shaft is determined as follows:

Let W = power absorbed per minute;

P = unbalanced pressure or weight in pounds, acting on the lever arm at a distance L;

L = length of lever arm in feet from center of shaft;

N = number of revolutions per minute;

V = velocity of a point in feet per minute at distance L, if arm were allowed to rotate at the speed of the shaft $= 2\pi L N$

$$\text{Since brake horse power} = \frac{P V}{33{,}000}$$

substituting for V,

$$\text{B. H. P. (brake horse power)} = \frac{2\pi L N P}{33{,}000} \quad \ldots\ldots\ldots\ldots \quad (1)$$

It should be noted that if $L = 33 \div 2\pi$ the equation becomes

$$\text{B. H. P.} = \frac{2\pi}{33{,}000} \times \frac{33}{2\pi} \times N P = \frac{NP}{1{,}000} \quad \ldots\ldots\ldots\ldots \quad (2)$$

Accordingly, in order to use the simplified formula (2) the arm L is made $33 \div 2\pi$ or 5.285 feet, very approximately 5 ft. 3 7/16 inches.

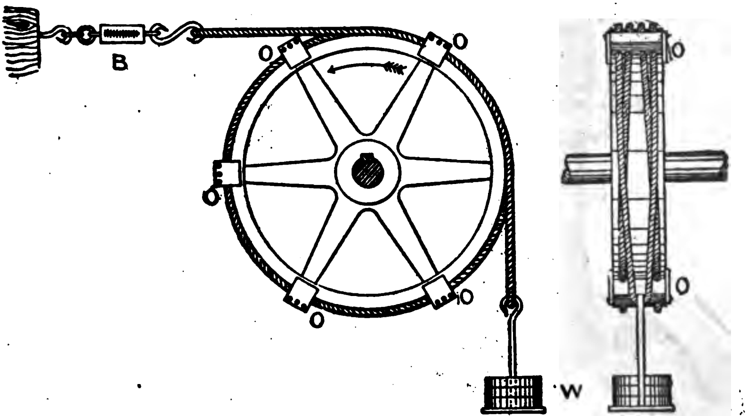
Ques. What important precaution should be taken in making a brake test?

Ans. The lever arm L, should be horizontal when a weight is used so that the force due to gravity *will act at right angles to the arm.*

If the arm be in any other position, the effect will be the same as shortening the arm and will introduce an error in the calculation.

Size of Cylinder.—Having learned the principles and methods of calculating the horse power of an engine, as given in the preceding pages, the student should now consider how to calculate the diameter and stroke of an engine to develop a given horse power.

It must be evident that for, say, a given speed, a great many cylinder sizes could be used, each giving the same power, that is, a long stroke with small piston, or a large piston with small stroke could be used, the best proportion between the stroke and diameter being determined by the type of engine, service for which it is intended, etc., and a knowledge of best practice on the part of the designer.



FIGS. 96 and 97.—Side and end view of rope brake. This type of brake is easily constructed of material at hand and being self-adjusting needs no accurate fitting. For large powers the number of ropes may be increased. It is considered a most convenient and reliable brake. In the figure the spring balance, B, is shown in a horizontal position. This is not necessary; if convenient the vertical position may be used. The ropes are held to the pulley or fly wheel face by blocks of wood, O. The weight at W, may be replaced by a spring balance if desirable. To calculate the brake horse power, subtract the pull registered by the spring balance, B, from the weight W. The lever arm is the radius of the pulley plus one-half the diameter of the rope. The formula is,

$$*\text{B. H. P.} = \frac{2\pi R N (W - B)}{33{,}000}$$
$$= .0001904\ R N (W - B)$$

In the formula R = radius from center of shaft to center of rope; N = *revolutions* per minute; W = weight; B = spring balance.

For a given piston speed evidently a short stroke engine will make a larger number of revolutions per minute than one with a long stroke. Thus, for say, 800 ft. of piston speed, an engine with 1 ft. stroke will make 800 ÷ (2 × 1) = 400 r.p.m., whereas, for 2 ft. stroke only 800 ÷ (2 × 2) = 200 r.p.m. will be required.

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*NOTE.—If B be greater than W, the engine is running in the opposite direction; in this case use the formula B. H. P. = .0001904 R N (B — W).

Ques. What is the meaning of the term high speed engine?

Ans. It means that the number of revolutions per minute or *rotary speed* is high, for the particular type of engine in question. This term should not be confused with high piston speed, as it does not relate to the piston speed.

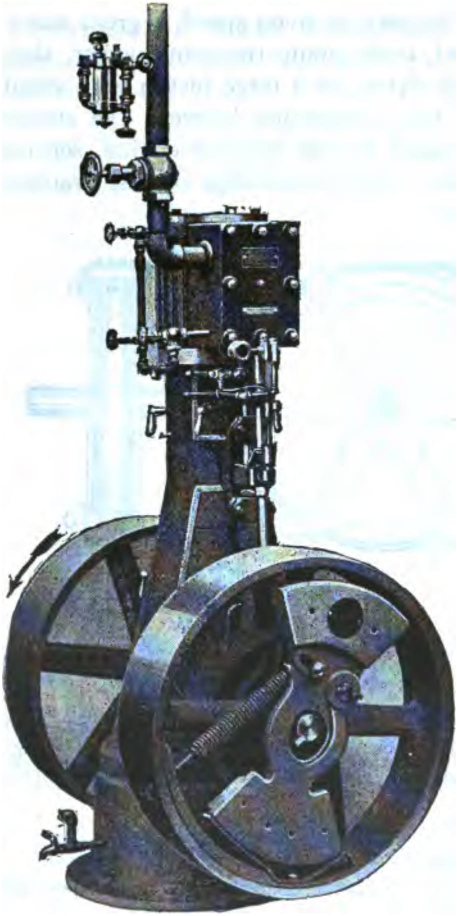

FIG. 98.—Troy high speed vertical automatic center crank engine with self-oiling system. Made in sizes ranging from 3¼ × 4 to 12 × 12; revolutions: highest 600 to 350, standard 400 to 300; lowest, 400 to 275. The oiling system consists of a reservoir in base for oil, a pump driven from the eccentric rod, and pipe connections to all the bearings. ***In operation*** the oil is drawn from the supply in the engine base, through a strainer funnel and suction pipe to the pump and check valve, then driven through the sight feed, where its movement can be noted, to the distributing head and thence through the supply pipes to the bearings, keeping them flooded; and, overflowing, finds it way back to the reservoir for repeated use. Any water of condensation entering the reservoir is automatically carried away, leaving the oil. No water can enter the suction pipe if the designated amount of oil be placed in the engine. A special packing is used in the piston rod stuffing box and practically eliminates the passage of water.

The first step in calculating the cylinder dimension is to arrange the horse power formula in the proper form for obtaining the value of the unknown quantity.

Thus, starting with the formula

$$\text{H. P.} = .000004\, D^2\, L\, N\, P \quad \ldots\ldots\ldots\ldots \quad (1)$$

the quantities to be found are D, the diameter of cylinder or piston, and L, the length of stroke. Accordingly, solving for these quantities

$$D^2 = \frac{\text{H. P.}}{.000004\, L\, N\, P}, \text{ or } D = \sqrt{\frac{\text{H. P.}}{.000004\, L\, N\, P}} \quad \ldots \quad (2)$$

$$L = \frac{\text{H. P.}}{.000004\, D^2\, N\, P} \quad \ldots\ldots\ldots\ldots\ldots \quad (3)$$

For those who do not understand the solution of equations, (2) and (3) are easily obtained from (1) as follows:

Rule.—*On one side the equality sign write down the unknown quantity; on the other side,* 1, *the horse power as numerator, and* 2, *the remaining factors as denominator.*

Of course, when D is the unknown, since it is squared in the formula, the square root must be taken as in (2).

Example.—Find the size of cylinder of a Corliss engine to develop 85 horse power when running under the following conditions: Initial pressure, 80 lbs.; ¼ cut off; mean back pressure, 2 lbs. (non-condensing) diagram factor .9; piston speed 600 ft. per minute.

The solution consists of three steps, viz.: finding, 1, the mean effective pressure; 2, the stroke, and 3, the diameter of cylinder.

CASE 1. DIAGRAM FACTOR GIVEN

1. Mean effective pressure.

1. Find total number of expansions (neglecting clearance).†

Rule.—One divided by the reciprocal* of the cut off.

$$1 + \frac{1}{\frac{1}{4}} = 1 \times 4 = 4$$

2. Find mean forward pressure.

***Rule.**—Multiply initial pressure by* $1 + hyp.\ log.$ *of expansions, and divide by number of expansions.*

From table page 71, hyp. log. of $4 = 1.3863$.

$$1 + \text{hyp. log. } 4 = 1 + 1.3863 = 2.3863$$

$$\text{initial pressure absolute} = 80 + 14.7 = 94.7$$

$$\text{mean forward pressure} = \frac{94.7 \times 2.3863}{4} = 56.5 \text{ lbs. per sq. in.}$$

3. Find mean effective pressure.

***Rule.**—Subtract mean back pressure absolute from mean forward pressure, and multiply the difference by the diagram factor.*

2 lbs. mean back (gauge) pressure $= 2 + 14.7 = 16.7$ lbs. absolute

$$(56.5 - 16.7) \times .9 = 35.8 \text{ lbs. per sq. in.}$$

2. Choice of Stroke.

The length of stroke must be such as will give a desirable number of revolutions, and bear a proper relation to the cylinder diameter. The Corliss engine is a slow speed or long stroke type, usual ratio of stroke to diameter being about 2 : 1 or more, hence of the several lengths of stroke that could be used, one should be selected that will come within the ratio limits and also give the proper speed in developing the rated power. Ordinarily the revolutions may be from 100 to 125, and with valve gears

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†NOTE.—It should be understood that in the example the expression one-quarter cut off relates to the point of stroke at which steam is cut off by the valve gear; it does not represent the *real* cut off, with respect to the expansion of steam, because clearance must be considered, and on this account is, strictly speaking, called the *apparent* cut off, which will be explained in the chapter on valve gears. The economical range of horse power being considerable, correction for the apparent cut off need not ordinarily be made.

*NOTE.—The *reciprocal* of the cut off means one divided by the cut off.

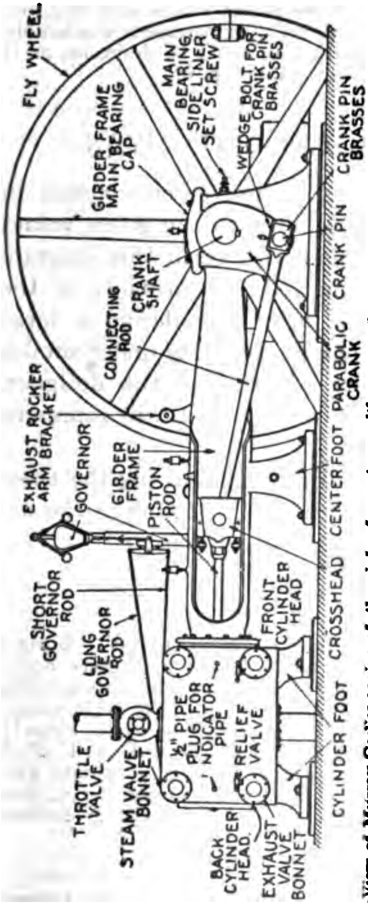

FIG. 99.—View of Murray Corliss engine of the girder frame type with names of parts and giving a general idea of the appearance of the engine considered in the accompanying example.

especially designed for high speed, 150 r. p. m., or higher. The revolutions or

$$\text{R. P. M.} = \text{piston speed} \div 2 \times \text{stroke (in feet)}$$

thus, for say, 24" stroke and given piston speed of 600 feet

$$\text{R. P. M.} = 600 \div 2 \times \frac{24}{12} = 150$$

Similarly, the following table is obtained:

R. P. M. for 600 ft. piston speed

Stroke	24	30	36
R. P. M.	150	120	100

3. *Diameter of cylinder.*

The m. e. p. obtained in *1*, is 35.8 lbs. per sq. ins.; now inspecting the table in *2*, a trial may be made with the 36" stroke which gives 100 r. p. m. Substituting the values in formula (2) page 97,

$$D = \sqrt{\frac{85}{.000004 \times 36 \times 100 \times 35.8}} = 12.8 \quad . \; . \; . \; . \; . \; . \; . \; (a)$$

For the given power, this diameter of cylinder may be used with any stroke in the table in *2* at the revolutions given, that is the cylinder dimension may be

12.8×36 for 100 r. p. m.
12.8×30 for 120 r. p. m.
12.8×24 for 150 r. p. m.

calling the diameter 13 ins., in each case, the stroke diameter ratios are 2.77, 2.3, and 1.87 respectively, the first two being within limits and the last two small.

In the case of a growing plant where more power will be soon required the 13×36 would be desirable, as the r. p. m. could be increased considerably to increase the power. Ordinarily, the 13×30 would be desirable, as it would cost less, and would run at a more desirable r. p. m.

CASE 2. DIAGRAM FACTOR NOT GIVEN

A graphical solution of the example just given consists in drawing the theoretical card corresponding to the given values of initial pressure, cut off, etc., and inscribing in this diagram a card drawn to represent the "expected" performance of the actual engine. This card is drawn after considering a large number of actual cards of similar engines operating under similar conditions. Accordingly, the more experienced the designer, the nearer can he come to drawing a card that will represent the actual performance of the engine.

The steps in this graphical method are: 1, drawing the theoretical card; 2, drawing the expected card; 3, finding the expected m. e. p.; 4, finding the cylinder dimensions.

1. The theoretical card.

In fig. 100, draw the vacuum line, or line of no pressure OV. Using a scale of 1″=40 lbs., draw the atmospheric line E D, a distance above corresponding to 14.7 lbs. and parallel to O V. At a height corresponding to 94.2 lbs. abs. draw the admission line A B, in length = ¼ of E D; extend A B, by dotted line to 3 and at points 1, 2, 3, drop perpendiculars. From O, draw radial lines 01, 02, 03, cutting the perpendicular from B, at 1′, 2′, 3′ respectively; the intersection of horizontal lines from these points with the perpendiculars, give points 1″, 2″, C, on the expansion curve, thus completing the theoretical card A B C D E, corresponding to the given data.

2. The expected card.

At this point all depends on the experience and judgment of the designer who sketches within the theoretical card, fig. 100, a card which he thinks will represent the actual performance of the engine. He proceeds about as follows: The initial pressure being given instead of boiler pressure, no allowance is made for drop between boiler and engine, hence the *expected* admission line begins from A, and proceeds toward B, first horizontally, and then begins to slope downward (though very slightly) because of pressure drop due to *initial condensation* due to the low temperature of the cylinder walls; approaching cut off at *b*, the admission becomes curved because of drop due to "wire drawing" as the valve closes.*

The expected expansion line then begins at *b*, at a lower pressure than B. Because of this condition and the fact that condensation continues, part of the stroke, say to *c*, a point at which the temperature of the steam and cylinder walls are considered the same.

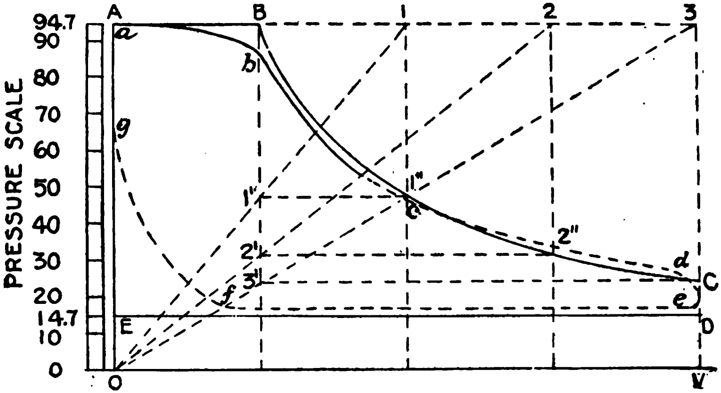

FIG. 100.—*The expected diagram.* Having drawn the theoretical card A B C D E, success in obtaining the proper diagram factor depends upon the experience and judgment of the designer which guides him in sketching in the expected diagram *a b c d e f g*, which he "expects" will represent the actual performance of the engine when built and operating under the specified conditions.

Expansion beyond *c*, will evidently take place at temperature lower than that of the cylinder walls, which is regarded as practically constant. Accordingly re-evaporation of some of the steam previously condensed will take place causing the expected curve to approach and rise above the theoretical curve between *c* and *d*.

At *d*, *pre-release* occurs, the gradual opening of the exhaust valve causes a pressure drop, represented by the rounded end or toe *d e*. Release is taken at 2 lbs., and represented by a line extending from *e*, to some point

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*NOTE.—In the Corliss engine this loss is reduced to a minimum owing to the very quick movement of the valve, but the act of cutting off steam in any valve gear is far from being instantaneous.

f, selected by the designer at which the exhaust valve closes, that is, at which compression begins.

The compression curve extending from f, to g, may be sketched in by eye, or if greater accuracy be desired, by constructing a hyperbolic curve, based on the clearance. The latter in the Corliss engine may be taken at 2½ to 3% of the piston displacement.

3. *The expected M. E. P.*

This is determined from the area of the expected card, fig. 100, by means of the following formula:

$$\textit{expected m. e. p.} = \frac{\textit{area of card}}{\textit{length of card}} \times \textit{pressure scale} \quad \ldots\ldots \quad (1)$$

The area is best obtained by use of a planimeter, or approximately by ordinates (explained in figure 81). In this case the area is by planimeter; length of card 4 ins., and pressure scale 40. Substituting these values in (1)

$$\textit{expected m. e. p.} = \frac{3.42}{4} \times 40 = 34.2 \text{ lbs. per sq. ins.}$$

4. *Cylinder dimensions.*

In Case I, a 36″ stroke was selected for future excess power demands, and a 30″ stroke where such provision was not made. Substituting the value 34.2 lbs. per sq. ins. *expected m. e. p.* in the formula for the 36 ins. stroke

$$D = \sqrt{\frac{85}{.000004 \times 36 \times 100 \times 34.2}} = 13.1 \text{ ins. say } 13 \text{ ins.}$$

As in Case I, the cylinder dimensions could be either 13×36, or 13×30 according to conditions and judgment of the designer.

CHAPTER 3

STEAM ENGINE PARTS

The numerous parts of which an engine is composed may be divided into three classes with respect to operation, as

1. Stationary;
2. Revolving;
3. Reciprocating.

The stationary parts are the cylinder, frame and bed plate; the revolving parts, the shaft, eccentric; the reciprocating parts, the piston, piston rod, crosshead, connecting rod, valve, and valve gear. Of these various parts the greatest proportion are reciprocating, and these, especially in the case of high speed engines, must be of minimum weight consistent with proper strength to avoid undue vibration, thus, the skill of the designer is shown by his treatment of these parts.

The Cylinder.—This consists of a cylindrical chamber, as shown in figs. 101, and 102, bored true, and in which is fitted a steam tight piston, free to move from one end to the other.

The distance in which the piston, during its stroke, is in contact with the cylinder is called the *bore*. To prevent a "shoulder" being formed at either end by the action of the piston, the diameter at these points is enlarged so that the piston slightly *overtravels* the bore. The enlarged sections are known as the *counter-bore*.

The cylinder is closed by two covers called *cylinder heads*. These are secured to the flanged ends or *faces* by bolts. All bearing surfaces are finished smooth and true and a steam tight joint is made at each face by

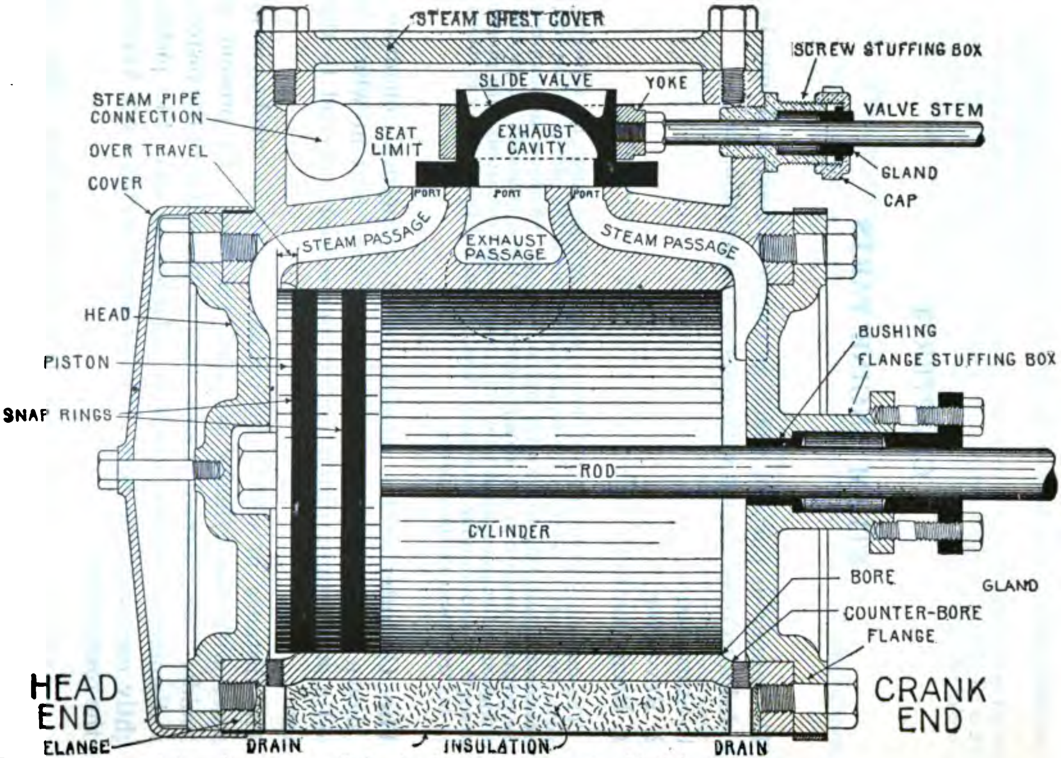

FIG. 101.—Sectional view of an engine cylinder showing the various parts with their names.

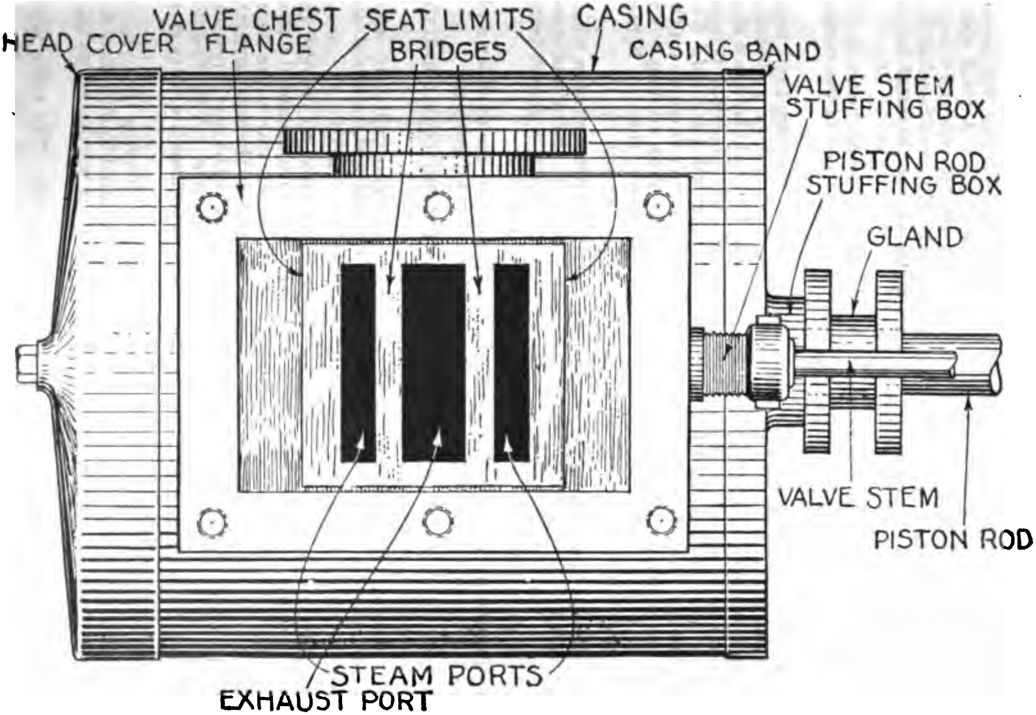

FIG. 102.—View of steam chest of the cylinder illustrated in fig. 101, with cover and valve removed, showing the valve seat, ports, etc.

inserting a *gasket*, that is, a thin sheet of packing cut to size, and then bolting the parts firmly together.

The distance between the faces of the cylinder is such that the piston does not touch either head when at the end of the stroke. A small space is always left between to prevent contact. This volume plus the volume of the steam passage between the cylinder and the valve seat is called the *clearance, and is expressed as a percentage of the volume displaced by the piston in one stroke.*

The term clearance is also used to denote the distance between the cylinder head and the piston when the latter is at either end of the stroke, being called the *linear clearance.*

A hole is bored through one head for the piston rod, a steam tight joint being made by means of a *stuffing box.* This consists of a cylindrical chamber, of somewhat larger diameter than the rod. The annular space thus left around the rod is filled with fibrous, or metallic *packing* which is compressed so as to form a tight joint by a hollow sleeve called a *gland.* The latter is forced into the stuffing box to bring the necessary pressure on the packing by adjusting the two bolts. In some cases a screw stuffing box is provided, similar to the one shown for the valve stem; this is a lighter construction but is liable to come unscrewed unless locked.

The cylinder has a projection on one side called the *steam chest* in which is the valve. The steam chest is closed by a plate known as the *valve* or *steam chest cover*; this is fastened to the steam chest by bolts, a gasket being placed between the bearing surfaces to make a tight joint.

In operation, steam passes from the steam chest to the cylinder ends through the *steam passages*, between which is the *exhaust passage.*

At the beginning of these passages is a smooth flat surface on which the valve moves and which is called the *valve seat.*

The two openings in the valve seat to steam passages are called the *steam ports*, and the opening to exhaust passage the *exhaust port.* Careful distinction should be made between the terms passages and ports.

The valve as shown overtravels the length of the seat to prevent the formation of a shoulder by wear, and also, in the case of unbalanced valves to reduce the load on the valve pressing it against its seat due to the steam pressure, as when the steam edge of the valve overtravels the seat limit, the pressure on that portion of the valve not in contact with the seat is neutralized. Motion is transmitted to the valve by the *valve stem.* A stuffing box is provided to make a steam tight joint at the point where the valve stem passes out of the steam chest.

Loss of heat by radiation is partially prevented by covering the cylinder with asbestos or other *insulating* material. For external appearance and to protect the insulating material, it is covered with a wooden or metallic lagging.

An interior view of a steam chest is shown in fig. 102. The valve being removed, the valve seat is exposed showing the steam and exhaust ports.

At the side of the cylinder is a projecting flanged pipe which forms the outlet from the exhaust passage.

The depression at the farther end should be noted; this terminates the valve seat and allows the valve to overtravel for reasons already explained.

There is a slight projection on the two side walls of the steam chest which is planed smooth to serve as a guide for the valve in its direction of travel. It also permits the valve being made narrower than the steam chest so that it may be easily inserted. These projections are sometimes omitted.

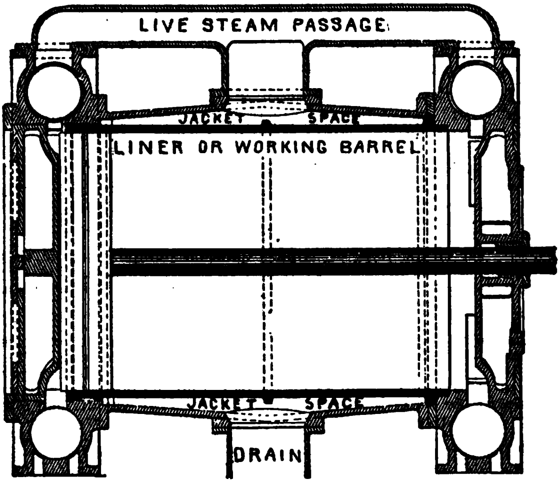

FIG. 103.—Harris-Corliss cylinder with steam jacket. The section in black is the liner or working barrel, around which live steam circulates. The object of a jacket is to reduce condensation within the barrel or cylinder proper.

Jacketed Cylinders.—Sometimes a *liner* or *working barrel* of somewhat smaller diameter is fitted to a cylinder as shown in fig. 103, leaving an annular space all around through which live steam circulates. The object of this is *to supply heat to the walls of the barrel, to make up for that abstracted during expansion and exhaust*, so that at admission, the walls will be as hot as possible, thus preventing condensation or reducing it to a minimum.

In general, *the greater the number of expansion, the greater the reduction of feed water consumption due to the use of a jacket.**

Prof. Schröoter from his work on the triple expansion engines at Augsburg, and from the results of his tests of the jacket efficiency on a small engine of the Sulzer type in his own laboratory concludes as follows: 1. The value of the jacket may vary within very wide limits, or even become negative; 2, the shorter the cut off, the greater the gain by the use of a jacket; 3, the use of higher pressure in the jacket than in the cylinder produces an advantage; 4, the high pressure cylinder may be left unjacketed without great loss, but the other should always be jacketed.

The usual method of fitting *liners* or *working barrels* to cylinders to form steam jackets are shown in figs. 104 to 106. The construction is such that the liner is free to expand or contract independently of the cylinder casting, a steam tight joint being made between the two by means of an ordinary stuffing box packed with fibrous packing.

For equal conditions, the gain by jacketing is greater for small cylinders than for large, because in small cylinders the cylinder surface per unit weight of steam passing through the engine is greater than in large cylinders. In some cases the jacket is so constructed that steam supply for the engine is used for the jacket, passing through the jacket on its way to the engine.

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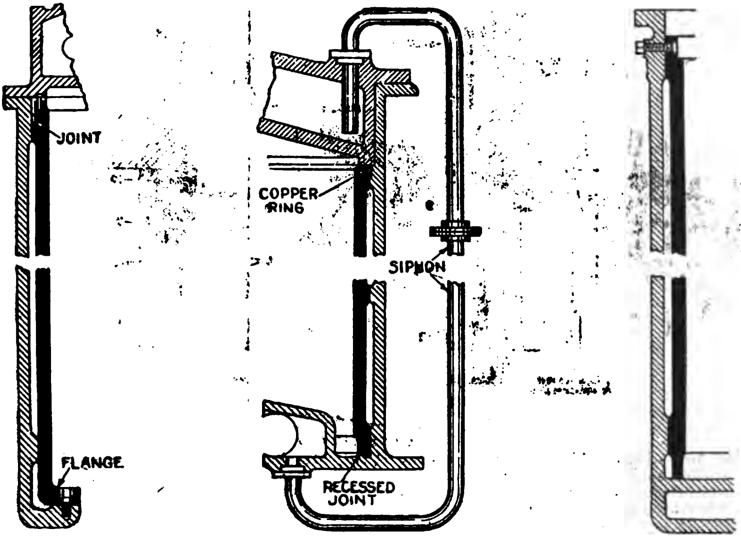
*NOTE.—A test of the Laketon triple expansion pumping engine showed a gain of 8.3 per cent by the use of the jackets, but Prof. Denton points out that all but 1.9 per cent of the gain was ascribable to the greater range of expansion used with the jackets (Trans. A. S. M. E., XIV, 1412).

NOTE.—***The value of the steam jacket*** may be judged from the experiments of Bryan Donkin made at Bermondsly on a single vertical experimental engine. The details of the engine are: Size 6 × 8; Meyer valve gear; barrel and heads jacketed, also valve chest cover, steam is supplied to each of the four jackets direct from the boiler by a separate pipe. By special arrangement the water from the cylinder body jacket was divided into two portions and the weight of them given separately. The first portion consisted of the steam condensed on the inner vertical surface of the jacket due to the heat passing through the walls into the cylinder; the other portion that condensed on the outer vertical surface of the jacket due to heat uselessly radiated outwards owing to imperfect external covering. Observations were taken with small thermometers inserted in one-eighth inch holes, drilled into the metal walls and filled with mercury. In all experiments the feed water always included the whole of the jacket water.

Mr. Donkin found that with about 50 lbs. boiler pressure, running condensing, the saving due to the jacket was: 40.4% at 6.8 expansions; 40.1% at 6 expan.; 38.5% at 4.8 expan.; 31.1% at 3.7 expan.; 23.1% at 1.8 expansions. In Mr. Donkin's experiments, the temperature of the cylinder itself was observed at various points between the inner and outer surface by means of thermometer inserted in small holes drilled in the metal. When the jackets were in use the mean temperature of the metal was almost equal to that of the steam on admission; when the jackets were not in use it was some 50° lower. The temperature as shown by the thermometer was nearly uniform from inside to outside; for the periodic chilling of the innermost layer of metal by re-evaporation of condensed steam was too superficial to be at all fully exhibited in this way. From the experiments it may be inferred that the smaller the cylinder, the greater is the percentage of gain from the use of a steam jacket arising doubtless from the fact that a small cylinder gives a larger jacket surface for a given weight of steam passing through it, than a larger cylinder does. (*2nd report of research committee on the value of the steam jacket.*)

Other engines experimented upon were: Compound jet condensing beam pumping engine; triple expansion pumping engine; compound mill engine. For full particulars of Donkin's experiments see *Pro. Inst. Mech. Eng.* 1892, page 464.

Stuffing Boxes.—By definition, a stuffing box is *a device affording passage and lengthwise or rotary motion of a piece, as of a piston rod or shaft, while maintaining a fluid tight joint about the moving part.*



FIGS. 104 to 106.—Various methods of fitting liners to cylinders. Fig. 104 is a form of liner having a flange at the bottom end secured to the cylinder by sunk head bolts. At the other end a steam tight joint is secured by a stuffing box with packing ring. Fig. 105 shows a liner with recessed joint at one end and at the other a plain contact joint reinforced by caulking a soft copper ring into a dovetailed groove. The figure shows a method of draining condensate from the top head by the pipe siphon. Fig. 106 shows a method of bolting the liner fast at the top with bolts spaced as far around the cylinder as the ports will admit. Although this method has been employed on U. S. Cruisers, the author does not consider it good practice. In fitting, the liner is forced into place but because of the danger of bringing too much strain upon the cylinder, the fit cannot be made tight enough to eliminate leakage, hence the necessity for stuffing boxes. The flanged joints at the bottom may be made tight with a gasket but usually a heavy coat of red lead or mastic cement is sufficient.

In construction, there is an annular space around the moving part, closed by an adjustable flanged bushing or gland so that when the annular

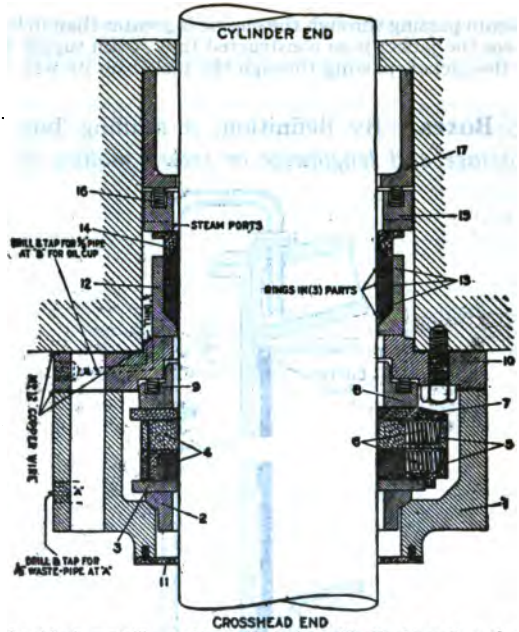
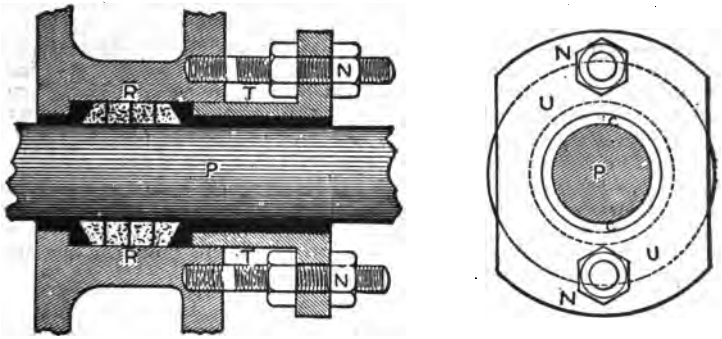
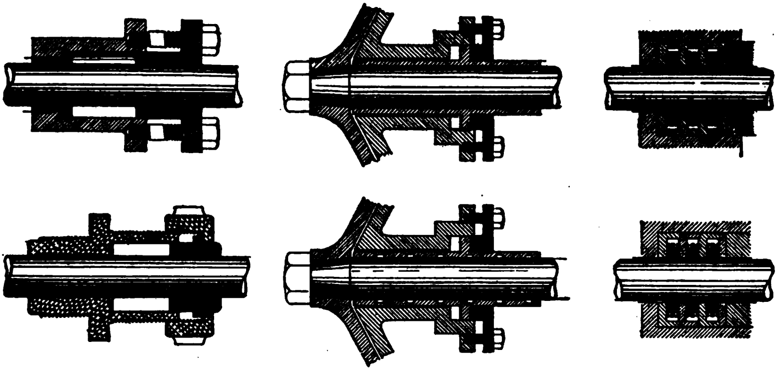


FIG. 107.—Duplex block metallic packing (U. S. Metallic Packing Co.), for stationary and marine engine piston rods and valve stems. This packing is in two independent sections, separated by the dividing piece 10, the upper section consisting of white metal rings 13, with vibrating cup, follower, etc., being in the stuffing box, and the lower section consisting of blocks 4, with horn rings, sliding plate, ball joint, etc., being contained in the gland 1. The upper or inner set of packings consists of three babbitt metal rings 13, each in three parts, contained in the vibrating cup 12. There is a ground joint between the flat face of the vibrating cup 12, and the dividing piece 10. Above the babbitt metal rings 13, come the follower 14, and the upper spring bushing 15, in pockets in which are contained the upper follower springs 16, and then in the bottom of the stuffing box the preventer 17, varying in length according to the depth of the box. As the dividing piece 10, is bolted to the face of the stuffing box independently of the gland 1, which contains the lower or outer set of packing, the blocks in the lower set may be renewed or any other necessary work done to same without taking down the upper set of packing or in any way disturbing it. The lower or outer section of the packing consists of eight blocks 4, babbitt lined, which are held in rings 5, having horns holding springs 6. The packing blocks are put together in sections four blocks to a section. Each section is composed of two working blocks and two guide blocks. The joints between the blocks in each section are at right angles to each other, thus "breaking joint." Small follower springs 9, in pockets are also used behind the follower plate 7. The combination of the sliding plates and ball ring 2, having ground surfaces, allows for the movement of rods out of line. The spring pressure is so regulated as to merely hold the parts in place when the engine is running without steam. A hole is drilled in the gland at A for a nipple and globe valve so that condensation can be drained off. *The parts are:* 1, gland; 2, ball ring; 3, sliding plate; 4, blocks; 5, horn rings; 6, horn ring springs; 7, follower plate; 8, spring bushing; 9, follower springs; 10, dividing piece; 11, swab-holder plate; 12, vibrating cup; 13, babbitt rings; 14, follower; 15, upper spring bushing; 16, upper follower springs; 17, preventer.

space is filled with fibrous packing the proper pressure may be applied to same to secure a tight joint.



Figs. 108 and 109.—Stuffing box which forms a steam tight joint for the piston or valve stem. By means of the adjustable sleeve, the proper pressure is brought to bear on the packing to prevent leakage of the steam.



Figs. 110 to 115.—Various types of stuffing box. Fig. 110 shows a plain box with stud adjustment; fig. 113, plain box with screw adjustment. Figs. 111 and 114 show the long sleeve type of box, these being identical except that in fig. 114 the sleeve has spaced grooves, as an extra provision against leakage In construction the sleeve is an accurate sliding fit with the rod and is secured by the flange which is held in a stuffing box, the latter not only serving to prevent leakage around the outside of the sleeve but to allow lateral movement of the sleeve to accommodate it to any irregularity in the movement of the rod. Figs. 112 and 115 show two types of metallic packing of the ring form. The rings have parallel faces and are held between the collars of a cast iron casing and the segments are pressed inward upon the rod by circumferential springs. In fig. 112, each ring is divided radially into three segments, and the two rings in one compartment break joint, being kept in place by little dowels which project into the gap in the elastic confining ring. The casing is divided lengthwise in halves.

In some cases the end surfaces of the annular chamber containing the packing are flat but usually are slightly conical to force the packing against the rod. The accompanying cuts show various types of stuffing box, and forms of packing used in same.

In design, the length and diameter of the stuffiing box depends on the material used and the working pressure. In the case of horizontal cylinders when the stuffing box becomes also a bearing, it may be made longer. For the valve stem, the box is proportionately deeper than for the piston rod.

In general, the stuffiing box may be from 2 to 3 times the diameter of the rod, and its diameter from 1⅝ to 1¾ times diameter of rod.

In some cases packing under pressure is dispensed with and a long plain sleeve depended on for a tight joint, the long close sliding fit between the rod and sleeve preventing leakage. The plain sleeve may be modified with several grooves spaced along its length to arrest the motion of any steam tending to leak through.

The Piston.—A piston may be described as *a device for receiving the pressure of, or operating upon, a liquid or gas in a cylinder or other enclosing vessel.*

Pistons may be classified:

1. With respect to shape, as

a. Cylindrical {solid; hollow;
b. Conical;
c. Rectangular.

2. With respect to motion, as

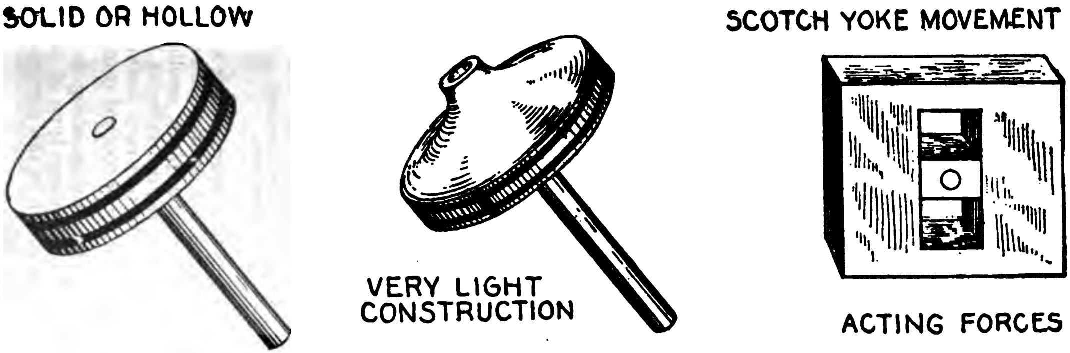
a. Reciprocating;
b. Oscillating;
c. Rotating.

3. With respect to the action of the steam as

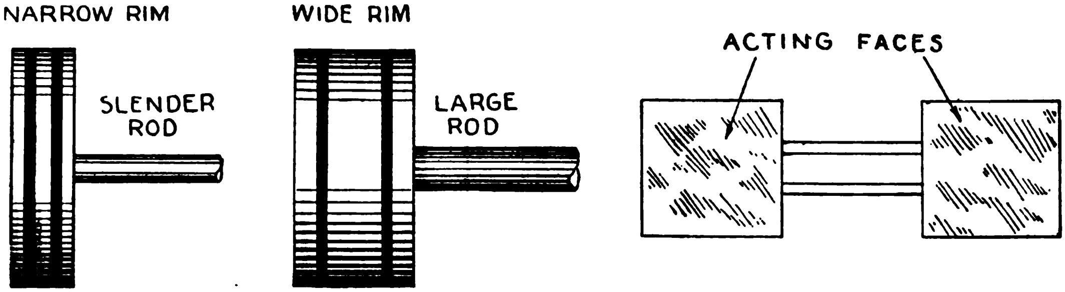
a. Single acting;
b. Double acting.

The cylindrical reciprocating piston is the most usual form and consists essentially of a disc attached to a rod and having rings which press against the cylinder walls to secure a steam tight joint.

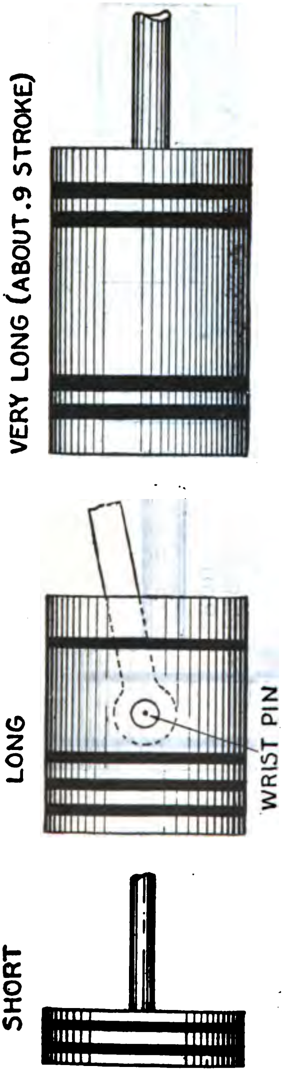
For high speed engines, especially those of the marine type, where the center of gravity of the engine must be a minimum, the piston is usually



FIGS. 116 to 118.—Various pistons classified according to shape. Fig. 116, cylindrical type; fig. 117, conical type; fig. 118, rectangular piston. The cylindrical piston is the most general type; the conical piston is used chiefly for marine engines because of its light weight; the rectangular piston is a type occasionally used in engines having rectangular or box like cylinders.



FIGS. 119 to 121.—Various pistons classified according to motion. Fig. 119, reciprocating piston; fig. 120, oscillating piston; fig. 121, rotating piston.



FIGS. 122 to 124.—Various pistons classified according to the action of the steam. Fig. 122, double acting piston; fig. 123, single acting piston; fig. 124, uni-flow piston. *The double acting piston* is short because it is not subjected to any lateral stresses of any magnitude. *The single acting piston* must be long because of the lateral thrusts due to the action of the connecting rod to which it is directly connected. Moreover, as there are no external stuffing boxes, it should have numerous rings to avoid any leakage of steam which in addition to being a loss would be a disagreeable feature in an engine room. *The "uni-flow" piston* belongs to the modern engine of that name, so called because the steam in passing through the cylinder does not reverse its direction. It is made 90 per cent. of the length of the stroke to permit the uncovering of ports in the cylinder walls when the piston is in the position corresponding to pre-release as later described in detail.

shaped like a cone, as this gives minimum weight, thus reducing vibration; its shape permits shortening the height of the engine because the stuffing box projects very little if any beyond the cylinder head.

A rectangular reciprocating piston consists of a square plate arranged to move to and fro in a rectangular box the steam pressure being received on the ends.

An oscillating piston is virtually the same as a reciprocating piston, but because of the lateral stresses, it is designed preferably with a wider rim giving more bearing surface to reduce wear.

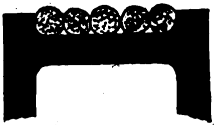


FIG. 125.—Early junk packing. This packing could neither be examined nor renewed without removing the piston from the cylinder. *To Remedy this,* the groove was made without a flange at the end farthest from the crank, and a false flange called a *junk ring* was bolted to the piston to retain the junk in place and admit of its being removed without taking out the piston. The term "junk ring" is still in use although junk is no longer used to pack pistons; a better term is *follower ring.*

Rotary engines have a rectangular rotating piston or rectangular plate oscillating in a sector cylinder and attached to a rack shaft or an eccentric or toothed cam.

The distinguishing feature of the various type of piston are shown in the accompanying cuts.

The three essential requirements of a piston are:

1. Strength to withstand the pressure of the steam.
2. A steam tight joint between its circumference and the cylinder walls;
3. Ability to move with very little friction.

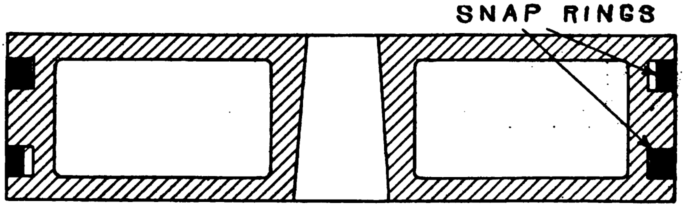


FIG. 126.—"Snap" piston rings. First used by Ramsbottom, an English engineer, and sometimes called "Ramsbottom's rings." They are turned somewhat larger in external diameter than the bore of the cylinder, and after being cut across so that they may be compressed to fit the cylinder bore, are fitted into recesses turned in the piston face.

According to Seaton, in the early days, owing to imperfect tools, cylinders were not bored true nor were the sides very smooth. Since the steam pressures at that time were quite low, pistons could be made steam tight by coiling rope or *junk* soaked in melted tallow in a groove on the rim of the piston as shown in fig. 125.

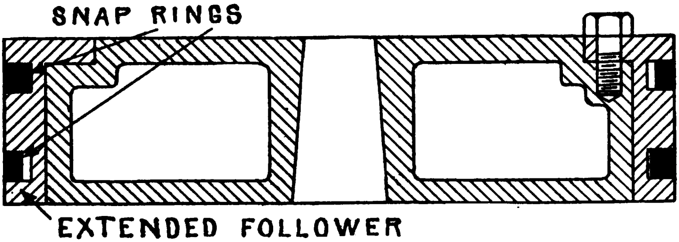


FIG. 127.—Snap rings fitted to extended follower. This permits removal of rings without taking out the piston.

The gradual increase in steam pressures soon caused the junk packing to give way to something more substantial.

Ramsbottom was the first to introduce metallic rings for piston packing. These rings, of small rectangular cross section, are turned somewhat larger in external diameter than the cylinder bore, and after being cut across, enough metal is filed off at the cut so as to compress them to fit the cylinder with the proper tension. The piston is turned to an easy fit and the rings fitted into recesses turned in its edge.

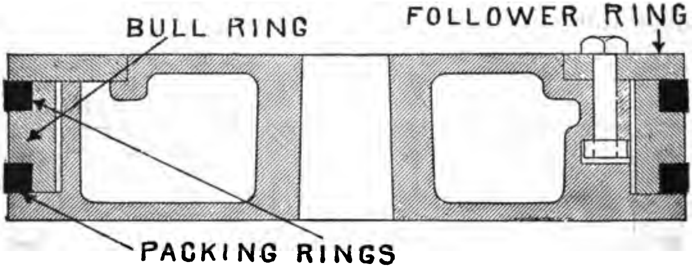
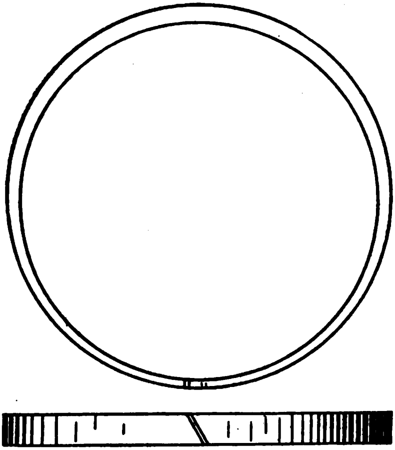
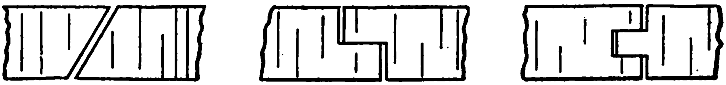


FIG. 128.—Snap rings fitted to *bull ring*. This is a modified form of the junk ring, the projecting spigot which carries the snap rings is cast separate as shown. The bull ring is held in place by a *follower plate*.



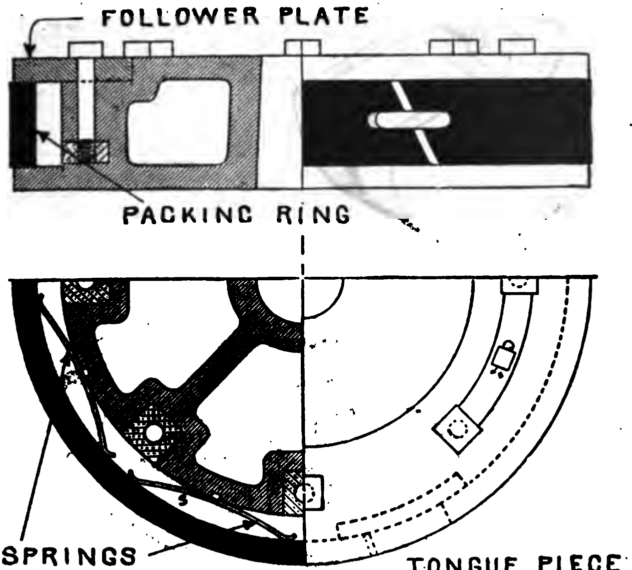
FIGS. 129 and 130.—Mode of turning snap rings to secure uniform pressure along the circumference. The ring is cut at the thinnest section.



FIGS. 131 to 133.—Different methods of cutting snap rings.

These rings are usually called ***snap rings***, since in putting them on the piston they "snap together" when they fall into the recesses.

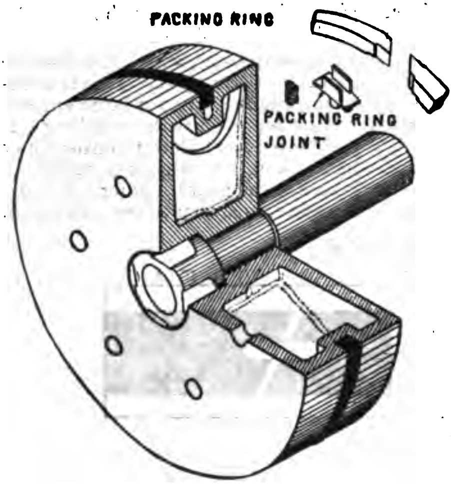
In fig. 126 is shown a piston fitted with two snap rings; when placed so the joints are not in line, the piston is practically steam tight. This is a desirable arrangement for small, quick running engines, but for large engines it has the objection that the rings cannot be removed without taking out the piston. This is overcome by fitting an extended follower consisting of a ring having cast with it a cylindrical extension, which goes down into a recess, and which is bolted to the outside face of the piston, snap rings being fitted into recesses turned on the outer circumference of the spigot as shown in fig. 127. By removing the bolts, the extended follower which carries the snap rings may be easily removed from the cylinder.



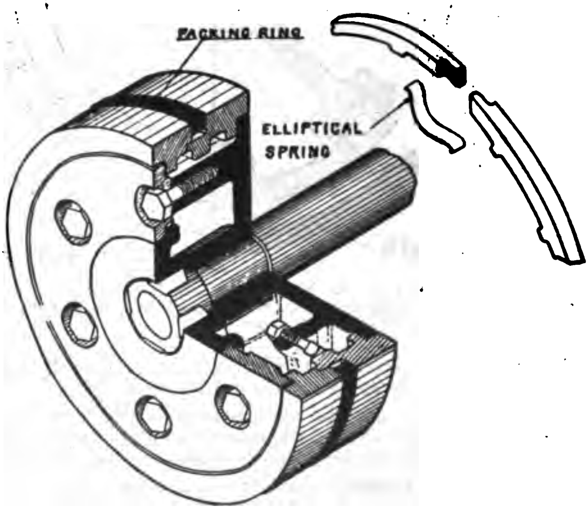
FIGS. 134 and 135.—Box piston. To reduce weight, large pistons are usually cast hollow with radial ribs. A single packing ring is shown pressed against the cylinder bore by the springs S. The ring joint is closed by a tongue piece T.

A modification of this arrangement consists of casting the extension separate. It is then called a *bull ring* and as shown in fig. 128 is held in place, together with the packing rings, by another ring called the *follower ring*.

Snap rings are usually made of tough, close grained cast iron, and turned eccentric as shown in fig. 129, being cut at the thinnest part. The object of



FIGS. 136 to 140.—Murray-Corliss hollow piston without follower.



FIGS. 141 to 144.—Murray-Corliss built up piston. The packing ring is carried on a junk ring, and retained in place by a follower and follower plate. By unscrewing the bolts, all these parts are easily taken out without removing the piston from the cylinder.

this is to cause the ring to press against the cylinder bore with uniform pressure at all points on the circumference.

The several ways of cutting snap rings are shown in figs. 131 to 133. Recesses for snap rings are turned deeper than the thickness of the rings to allow a transverse movement independent of the piston body, thus providing for lack of alignment between the piston and the cylinder.

In large cylinders, the necessary pressure of the ring against the cylinder is secured by means of a series of springs as shown in figs. 134 and 135. In this construction the packing ring usually consists of one large ring pressed outwards against the cylinder by springs, and retained in place by a follower plate.

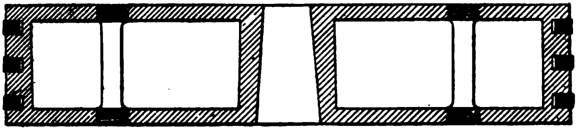


FIG. 145.—Box piston with wrought iron stay bolts instead of radial ribs, a lighter form of piston than that shown in figs. 134 and 135.

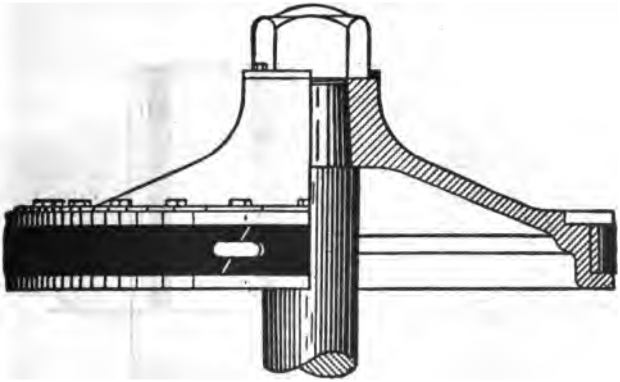


FIG. 146.—Cone piston as used on high speed marine engines. By making the piston in the form of a cone great strength is secured, thus reducing the weight.

For horizontal cylinders, the bottom spring is removed and a cast iron block put in its place, which takes the weight of the piston.

In construction, the body of a piston is either made:

1. Solid;
2. Hollow, or,

3. Built up.

For small engines, the solid type of piston is used, being simply a flat cast iron disc of sufficient thickness to receive the snap rings.

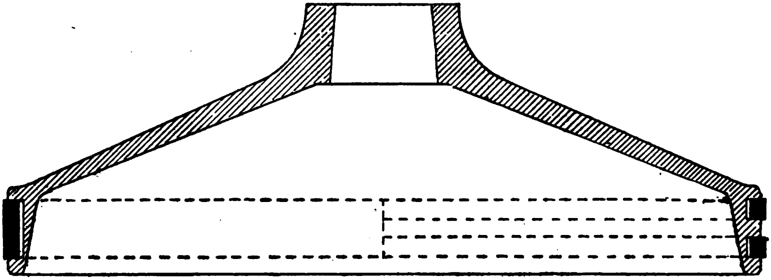
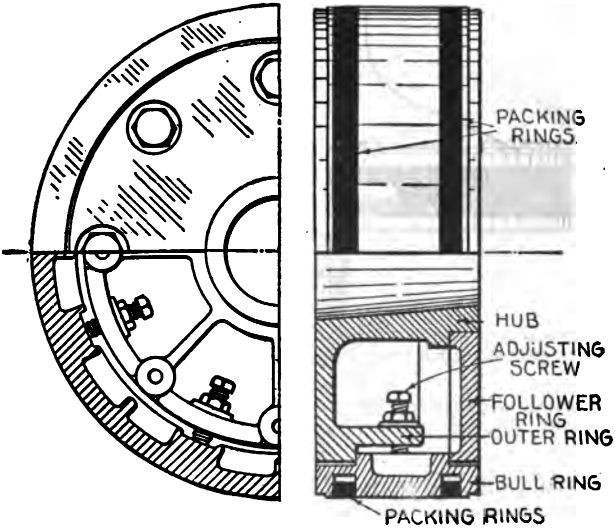
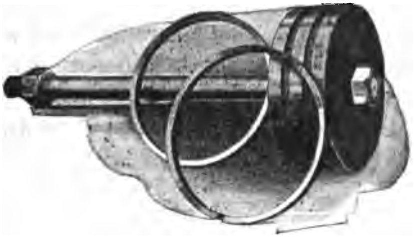


Fig. 147.—Cone piston of forged steel; very light construction and a desirable form for boats of high speed.



Figs. 148 and 149.—Built up piston consisting of spider and bull ring with its packing rings, and follower ring.

In the larger sizes, the piston is made hollow with strengthening ribs or reinforced by stay bolts to reduce the weight, as shown in figs. 134 and 135; this is called a *box piston.* Sometimes, instead of cast ribs, this form of piston is reinforced by wrought iron or steel stay bolts to further reduce the weight, as shown in fig. 145.



FIGS. 150 to 152.—Brownell piston, rings, and piston rod. The piston is of the solid type, cored and provided with two spring packing rings cut in such a manner that they make a steam tight joint. The piston is pressed on the rod by hydraulic pressure, and held in place by a nut.

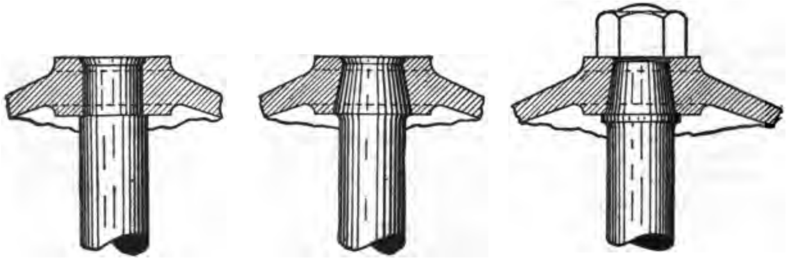
Where great strength and light weight are required, as in the case of high speed marine engines, pistons are made cone shaped, fig. 146, and for extreme light weight they are constructed of cast or forged steel, fig. 147, with one or more packing rings.

FIGS. 153 to 156.—Chandler and Taylor piston rings, piston rod crosshead and wrist pin. The piston is of the hollow cast iron type fitted with two snap rings. The piston rod is made of hammered crucible steel. It is fitted into the piston head on a taper, and is locked with a heavy nut. The piston rod is screwed into the crosshead, and is locked in position by a nut jammed tight against the boss on the crosshead.

The built-up type of piston is shown in figs. 148 and 149. It consists of: 1, a spider composed of several radial ribs, and a face cast with a central hub, and an outer ring; 2, a bull ring fitting over the outer ring and carrying the packing rings, and 3, a large follower ring enclosing the interior space of the piston and fastened to the spider by a number of bolts. The bull ring may be adjusted for proper alignment between the piston and the cylinder by means of the set screws.

The Piston Rod.—The load due to the steam pressure acting on the piston is transmitted by the piston rod out through the stuffing box to the crosshead and connecting rod. The alternate stresses of compression and tension which come upon the rod in rapid succession, severely test the material of which it is made, and since it is desirable that the rod be of small cross section, it is usually made of the best steel.

The piston rod has a uniform cylindrical shape except at the ends where it is joined to piston and crosshead. The rod should fit steam tight into the piston and be firmly secured to the latter so as to hold it rigid against shocks.



FIGS. 157 to 159.—Different methods of fastening the piston rod to the piston. Fig. 157, shrink fit with shoulder, end riveted; fig. 158, tapered joint secured by riveting; fig. 159 tapered joint with shoulder to prevent seizing, secured by a nut.

There are numerous ways in which the rod is fastened to the piston.

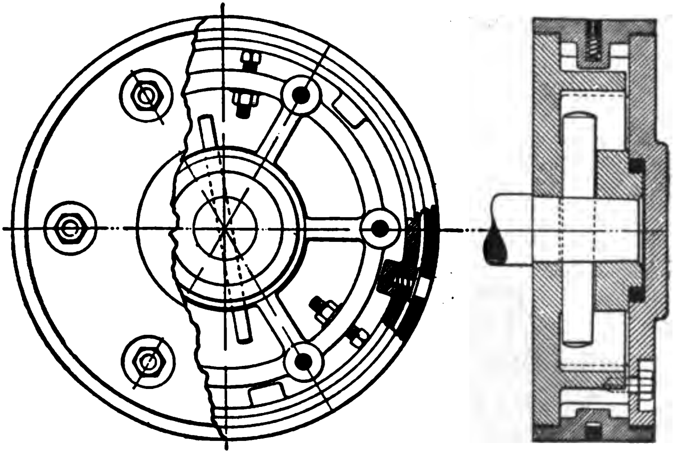
A simple method consists in reducing the diameter of the rod at the end, leaving a shoulder and making a shrink fit between the rod and piston; the rod is then riveted to the piston as shown in fig. 157. While this makes a cheap and firm joint, it is objectionable in that the rod cannot be readily removed.

To overcome this it is usual to make the end tapered or conical, as shown in fig. 158. If the taper be very slight the rod can be easily made a tight fit, but unless formed with a shoulder at the end of the taper, it will in time become so tightly held by the piston as to withstand all attempts at withdrawal, and there would be danger of splitting the piston by the wedging action.

In fig. 159 is shown a rod having a tapered end with a shoulder and secured to the piston by a nut. With the proper taper, this rod may be easily removed.

Piston rods are sometimes fastened to large built up pistons by means of a key, as shown in figs. 160 and 161.

The most usual method of securing the rod to the crosshead s a threaded joint and lock nut as shown in fig. 165. This has he advantage of permitting adjustment of the rod length so that the clearance spaces at the cylinder ends may be equalized with certainty.

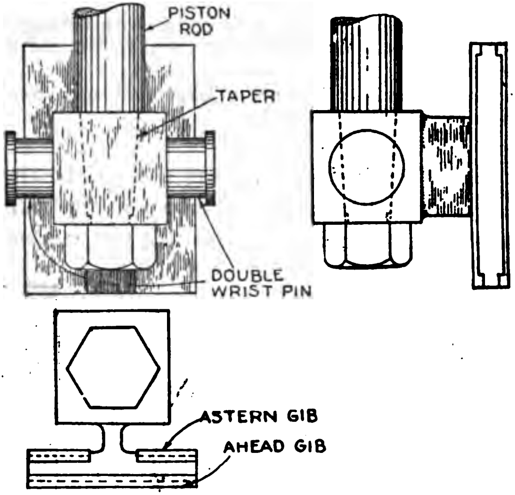


IGS. 160 and 161.—Large built up piston secured to the piston rod by means of a key.

The manner in which the rod is fitted to the crosshead depends somewhat on the type of the latter.

A form commonly used for marine engines is shown in figs. 162 to 164, consisting simply of a shoulder, taper, and a nut for holding the rod in position.

Another form used extensively in marine and other types of engines, consists in forging the body of the crosshead in one piece with the rod as shown in figs. 166 and 167. This construction has the serious objections,



Figs. 162 to 164.—Marine type of piston rod connection with crosshead, having a tapered joint secured by a nut. With this construction the height of the engine is reduced, lowering its center of gravity—a desirable feature for marine service.

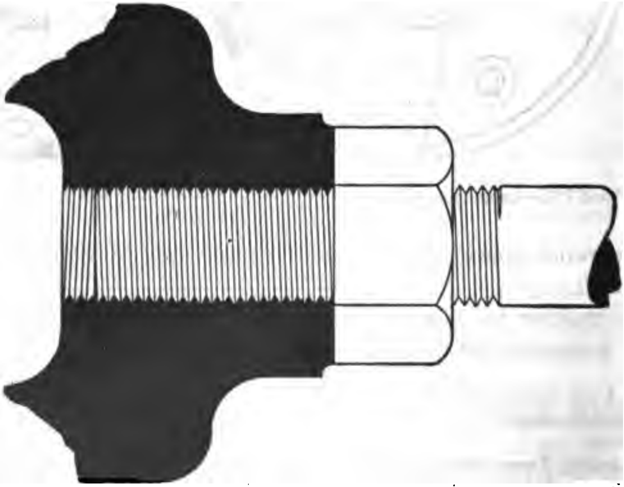
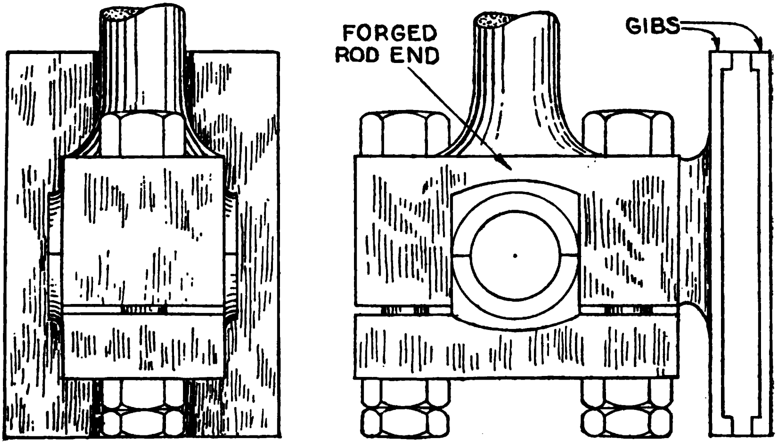


Fig. 165.—Usual method of securing the piston rod to the crosshead by a threaded joint with lock nut.

however, that if anything happen to the rod; repairs are not so easily nor quickly made as when the rod is a separate part, and the rod cannot be removed without disconnecting it from the piston.

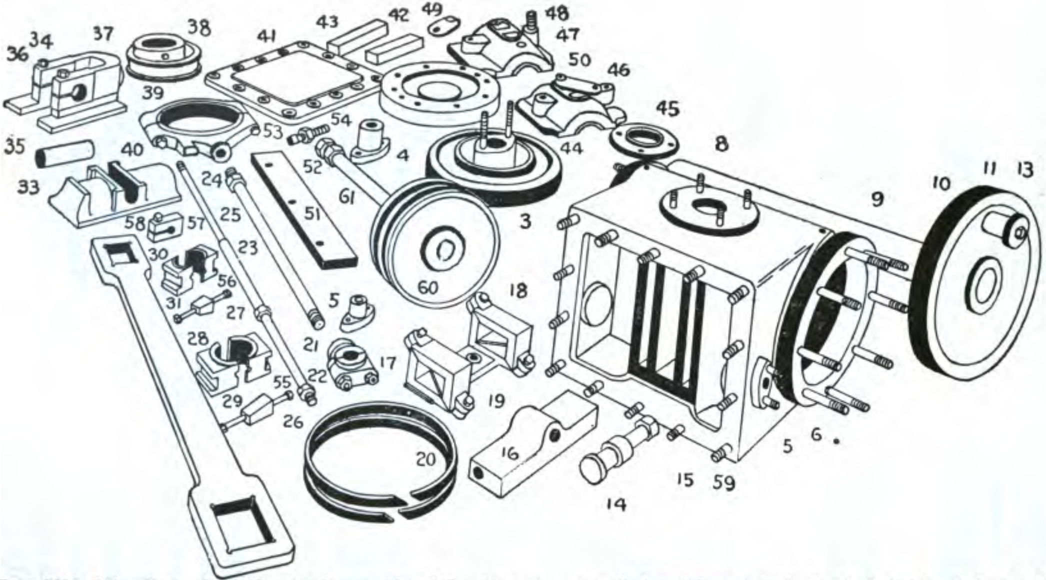
On locomotives the piston rod is usually secured to the crosshead by means of a key, figs. 160 and 161, the end of the rod fitting into a tapered hole in the crosshead.

When the engines are run intermittently and are idle for long periods, special care should be taken to protect the piston rod from corrosion and pitting. If a rod become pitted, it will be difficult to keep the stuffing box tight. When an engine is to remain idle for a length of time, the packing should be removed and the rod well oiled.



FIGS. 166 and 167.—Piston rod and crosshead forged in one piece; an objectionable though common construction.

To prevent corrosion, rods are sometimes made of phosphor bronze which has about the same strength as steel, and although more expensive is well worth the additional cost for engines which are to be run only occasionally.



FIGS. 168 to 228.—Parts of the Houston Stanwood and Gamble engine: 1, cylinder; 2, loose cylinder head; 3, tight cylinder head; 4, piston rod gland; 5, valve stem gland; 6, valve stem gland studs; 7, cylinder head studs—long; 8, governor flange studs; 9, main shaft; 10, crank disc; 11, crank pin; 12, crank pin cap; 13, crank pin cap screw; 14, valve bar pin; 15, valve bar pin nut; 16, valve bar; 17, valve bar guide; 18, valve bar guide cap; 19, valve bar guide cap bolts; 20, piston packing rings; 21, eccentric rod stub end; 22, eccentric rod stub end studs; 23, eccentric rod; 24, eccentric rod nut; 25, valve stem; 26, valve stem nut—threaded; 27, valve stem collar nut; 28, crank pin brass; 29, crank pin wedge; 30, crosshead pin brass; 31, crosshead pin wedge; 32, connecting rod; 33, slide valve; 34, cross head; 35, crosshead pin; 36, crosshead studs and nuts; 37 eccentric; 38, eccentric set screw; 39, eccentric strap; 40, eccentric strap bolts; 41, steam chest cap; 42, quarter box, short; 43, quarter box, long; 44, piston rod gland studs; 45, exhaust flange; 46, main bearing cap; 47, outboard bearing cap; 48, cap bolts; 49, oil cover; 50, oil cover screw; 51, slide bar; 52, piston rod nuts; 53, adjusting screws; 54, adjusting screw nuts; 55, crank pin wedge screws; 56, crosshead pin wedge screws; 57, valve stem lock nut; 58, valve stem lock nut clamp screw; 59 steam chest stud; 60 piston; 61, piston rod.

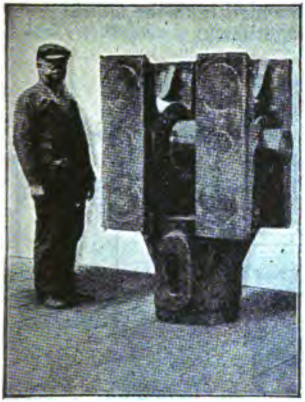
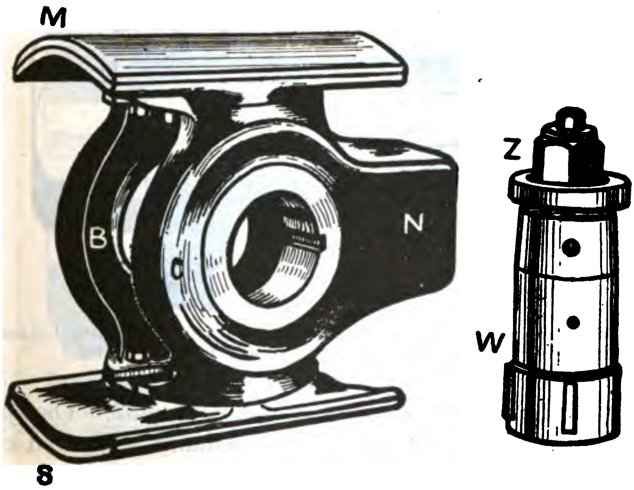


Fig. 229.—A large cross head.

The Cross Head and Guides.— A cross head is simply a "*sliding hinge*" *which joins the piston rod to the connecting rod.* By means of guides, it prevents the bending of the former which otherwise would occur on account of the side thrusts of the connecting rod.

The design of the cross head varies more than any other detail of an engine.

It consists essentially of a body, fig. 230, having two jaws B and C, between which the connecting rod is pivoted by the *wrist pin* W, fig. 231. This pin is inserted in holes bored in the crosshead body, and held firmly in place by a nut Z. At either side of the wrist pin are bearing surfaces M and S,



Figs. 230 and 231.—Cross head and wrist pin. The wrist pin W, is inserted in holes bored through the jaws B and C, and the pin is secured by the nut Z. M and S, are the gibs, which bear on the guides, and N, the neck to which the piston rod is fastened.

called gibs*. These run in suitable guides which take the side thrusts of the connecting rod. The two jaws come together in a *neck* N to which is attached the piston rod.

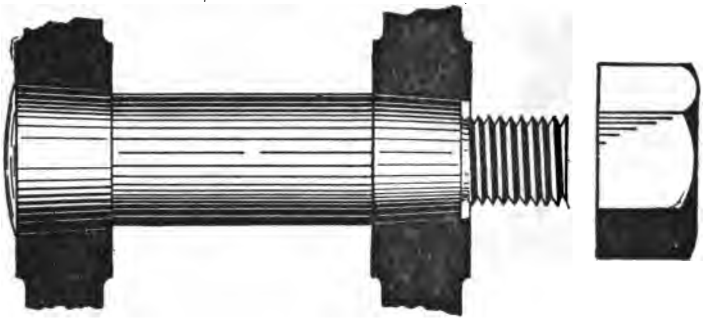


Fig. 232.—Wrist pin with tapered ends. The pin is drawn into very firm contact with the cross head by the nut on the end.

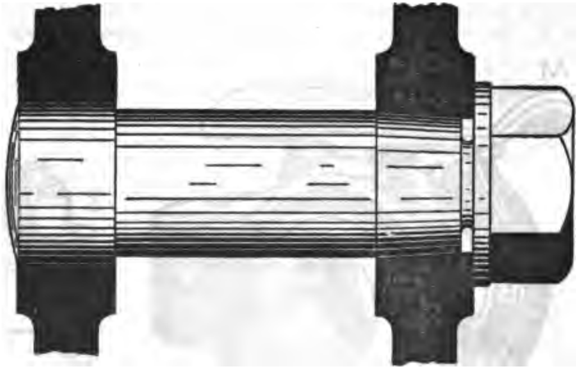


Fig. 233.—Wrist pin with cylindrical and tapered ends. A satisfactory method of attachment not requiring such precision in machining as when both ends are tapered.

The usual form of wrist pin is shown in fig. 232. That part in contact with the cross head body is usually a tapered surface which can be drawn into very firm contact by the nut on the end.

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*NOTE.—The cross head gibs are sometimes called shoes or slippers.

On loosening the nut, the pin is easily withdrawn. A key is usually inserted in the pin to prevent any turning.

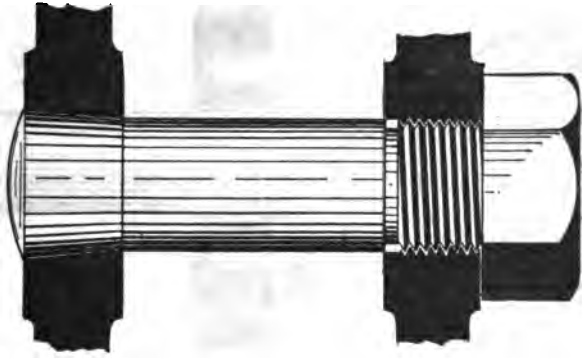


FIG. 234.—Wrist pin with tapered and threaded ends. This construction has the objection that the threads are liable to injury on account of the alternate transverse thrust to which they are subjected.

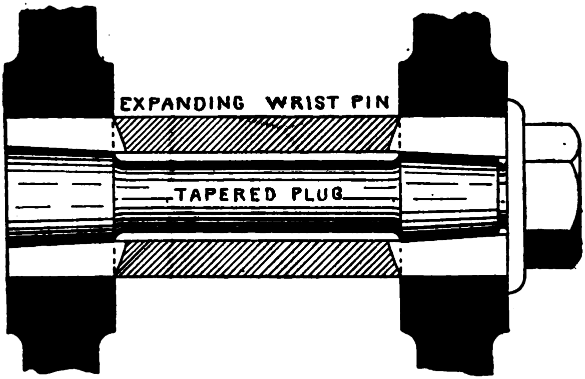
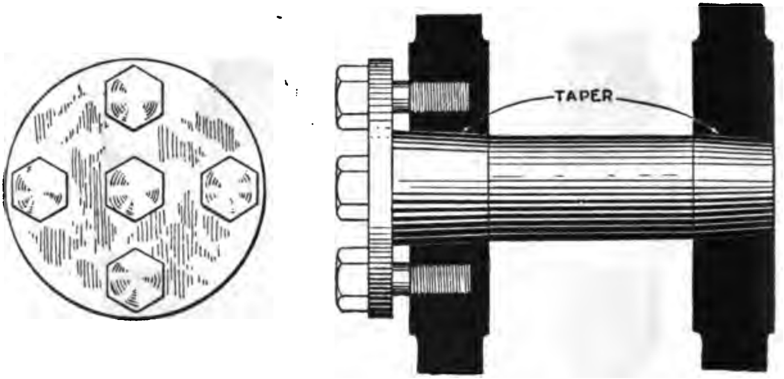


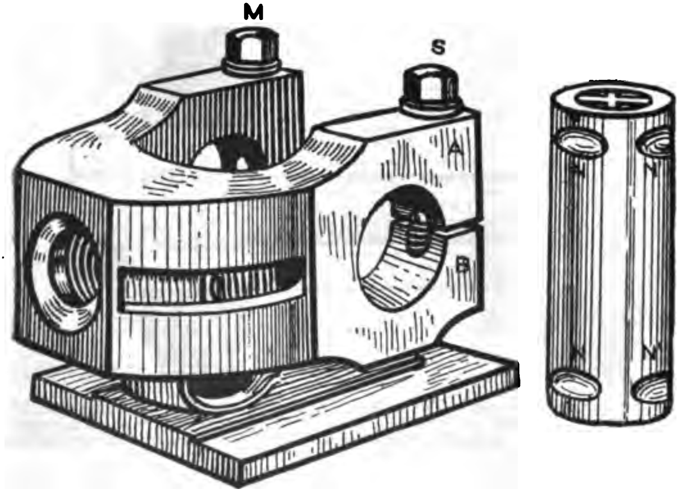
FIG. 235.—Expanding wrist pin. The ends are split and cylindrical, fitting accurately the holes in the crosshead jaws. An inner or expanding plug having tapered ends is drawn into the hollow wrist pin, thus expanding the ends, and firmly retaining the pin in position, an objectionable construction.

On some crossheads only one end of the wrist pin is tapered while the other is cylindrical as shown in fig. 233.

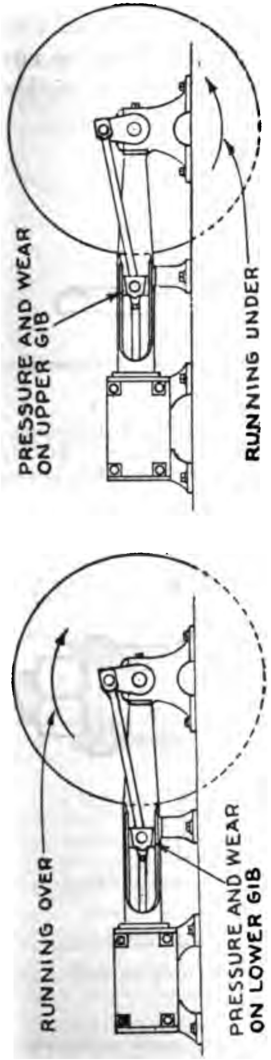
Another form of wrist pin is tapered at one end and threaded at the other, as shown in fig. 234, a lock nut being provided to retain the pin in position.



FIGS. 236 and 237.—Approved method of fastening wrist pin where only one side of the cross head is accessible. The pin which is tapered at both ends is held in place by bolts which pass through the flange.



FIGS. 238 and 239.—Cross head with compression wrist pin. The cross head jaws A. B. are split. and may be drawn together by the bolts M, S, which project slightly within the holes. These bolts register with notches N, N′ cut in the wrist pin. When the pin is in position the jaws are drawn together by bolts, tightly gripping the pin.


PRESSURE AND WEAR ON UPPER GIB
RUNNING UNDER
RUNNING OVER
PRESSURE AND WEAR ON LOWER GIB

Figs. 240 and 241.—Diagrams illustrating the terms "running over" and "running under" used to indicate the direction of rotation of an engine. Since the weight of the crosshead and connecting rod continually cause the lower gib to remain in contact with the guide when the wear is not fully taken up, if the engine run under the crosshead is suddenly thrown against the upper guide after passing the center, as in fig. 241, tending to cause a knock, which if the engine be run over would be avoided.

Some forms of wrist pin have no taper at the ends. In this class belongs the *expanding pin* as shown in fig. 235. The wrist pin is bored, and the bore tapered and split at each end where it rests in the crosshead To the taper in the pin is fitted a steel plug. In attaching the pin to the crosshead, the tapered plug, by means of a finely threaded end and nut, is drawn in, expanding the ends of the wrist pin against the sides of the crosshead This type of pin is used to advantage when only one side on the crosshead is accessible. However, the author's experience with this pin is that unless it be a very close fit with the crosshead it will work loose after being expanded by the plug, hence, it is not to be recommended.

A better method of fastening the wrist pin where only one side of the crosshead is accessible is shown in figs. 236 and 237.

A form of wrist pin known as the *compression type* is shown in figs. 238 and 239; the ends of the pin are without taper. The pin which accurately fits the holes in the crosshead jaws is notched at each end to conform to the bolts which project within each hole. Each jaw is split as shown, hence, when the pin is in position the bolts are tightened which causes the jaws to firmly grip the pin. Several notches are provided at the ends of the pin so its position may be changed from time to time to prevent the pin becoming flattened on account of wear.

The gibs of a crosshead may number one, two or four, but liberal bearing surfaces are provided on account of the velocity with which they move.

When a horizontal engine **runs over,** *all the pressure and wear comes upon the lower slipper,* as in fig. 240; *if the engine* **run under,** *all the pressure and wear comes upon the upper slipper,* as in fig. 241.

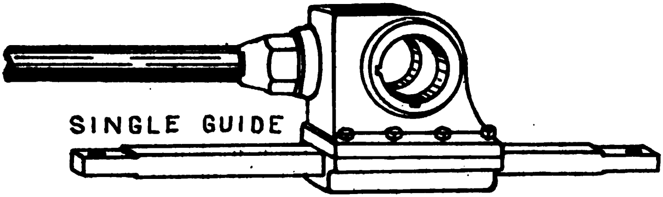
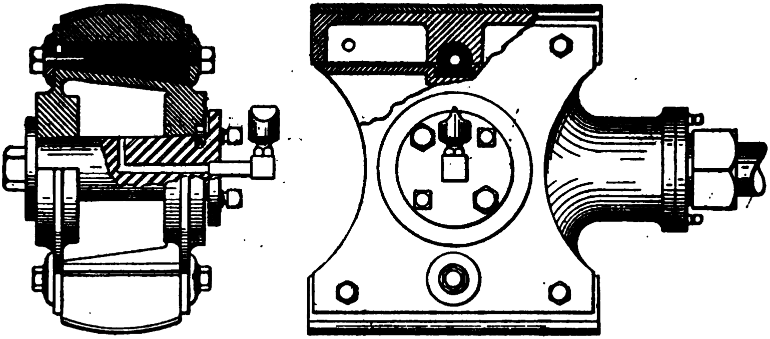


Fig. 242.—Cross head of the Reeves engine. A single bar machined on all four sides serves as the guides. This type is frequently used on marine engines.

Since lubricants flow over the lower guide more easily than the upper, and since it is easier to resist the strain on the lower guide, as it usually rests on the bed plate, it is customary to cause horizontal engines to run *over* rather than *under*.



Figs. 243 and 244.—Twin City Corliss cross head. Cylindrical gibs with cross wedge adjustment.

The greatest pressure upon a guide occurs when the piston is near the middle of its stroke which gradually diminishes in intensity to zero at the end of the stroke.

The pressure on the guide being, therefore, very small near the ends of the stroke it is not necessary that the entire surface of the gib be in contact with the guide at these points.[*] Hence, the guides may be shortened without harm and the gib be allowed to *overtravel* to quite an extent as shown in fig. 245. This is done to advantage in engines of very light weight or where parts may be made more accessible. In any case, the gib should overtravel the guide to prevent wearing a shoulder at the stroke ends.[†]

The guides may be one, two or four in number. In marine engines quite frequently only one is used.

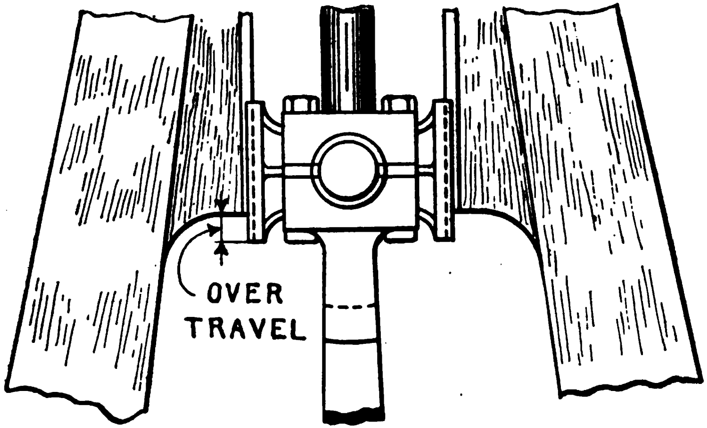


FIG. 245.—Detail of marine crosshead and guides illustrating overtravel. The gibs may project considerably beyond the end of the guide at the stroke ends, because the thrust on the guide diminishes to zero at the dead centers. The gibs should overtravel because a full length guide is unnecessary, and the parts are usually made more accessible by the shortened guides.

In this case there are two constructions: 1. In which the guide is surrounded by the crosshead as shown in fig. 242; 2. In which the crosshead projection containing the rubbing surfaces is partially surrounded by the guide as shown in figs. 246 and 247.

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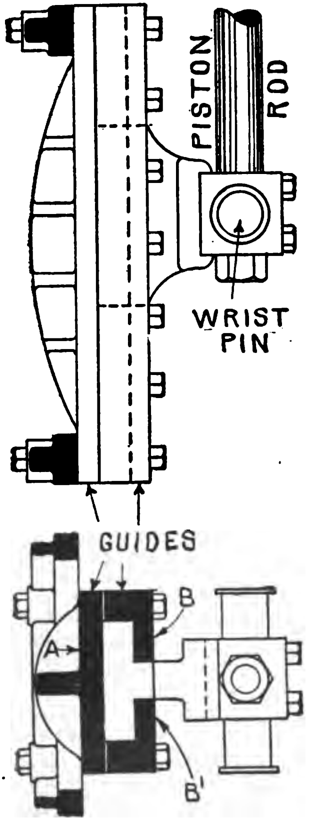
[*] NOTE.—The side thrust on the guide at any point of the stroke is obtained from the formula: side thrust = total load on the piston multiplied by the tangent of the angle which the connecting rod makes with the line of the piston rod $= p \tan\theta$. When the connecting rod is 2½ times the length of the stroke (the usual proportion), the maximum angle of the connecting rod with the line of piston is 11° 33′ and the tangent of this angle is .204 or approximately .2, hence, the greatest side thrust on the guide is .2 or 20 per cent. of the maximum load on the piston. For a 2:1 connecting rod, $\tan\theta = .258$; for 3:1 rod, $\tan\theta = .169$.

[†] NOTE.—The pressure between the crosshead slipper and the guide should not exceed 100 pounds per square inch of slipper surface. On many engines it is much less. In locomotives the pressure ranges between 40 and 50 pounds on account of dirt, cinders, etc.

This type is in fact two guides in one, and is peculiarly suited to marine engines since the area of the backing guides B, B′, need not be as great as that of the forward guide A. As a marine engine is run in the forward direction most of the time the backing guide may be of small area without harm, thus saving in weight and making the parts more accessible.

Cross heads for marine engines are sometimes made without a wrist pin and carry instead, the wrist pin bearing, the pin in the construction forming a part of the connecting rod.



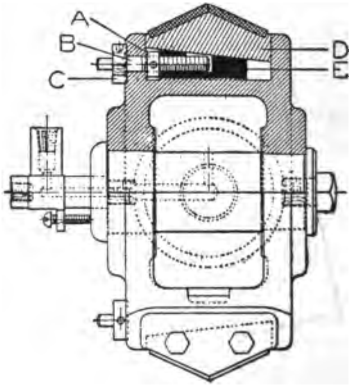
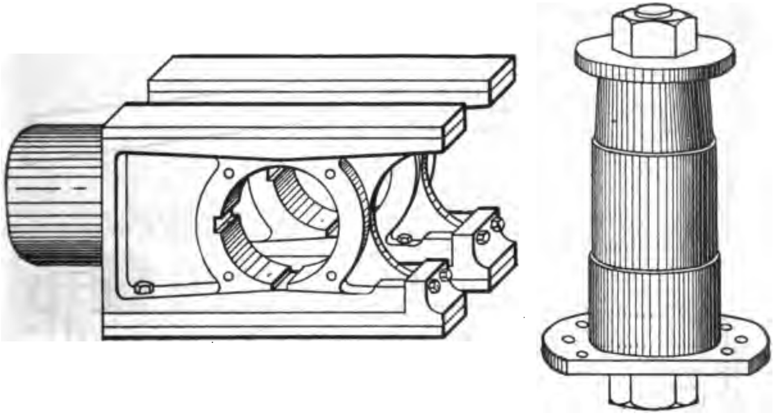


FIG. 248.—Cross head of the Brown engine. It is of heavy design, with large wearing surfaces. The cross head pin is placed in the center of cross head so that there is no tendency towards a rocking motion. The gibs, which are Babbitt lined, are keyed up by a wedge and screw from the face of crosshead. This construction allows the removal of the gibs without taking the cross head from the guides. In adjusting the cross head, loosen check nut C, and turn screw B, to the left to drive wedge E, in and force the gibs D, out to the right to bring the gibs in. Thrust collar A, is pinned to adjusting screw B.

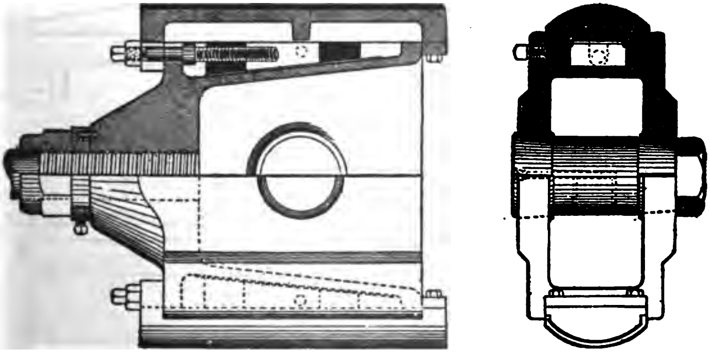
FIGS. 246 and 247.—Marine type of cross head and guides. The backing guides B, B′, present less area than the forward guide A, a condition well adapted to marine service because the allowable thrust on the guide is greater in backing than when going ahead, since the engine runs only for short periods when reversed. More liberal surface should be allowed for the surfaces B, B. in the case of ferry boat engines, because this type of vessel runs equal periods in both directions.

A marine crosshead of this type is shown in figs. 166 and 167. This is formed on the end of the connecting rod and is objectional in some respects as previously mentioned.

Stationary engines sometimes have crossheads designed for four guides, as shown in figs. 249 and 250, which is a view of the Porter-Allen crosshead and wrist pin.



Figs. 249 and 250.—Cross head and wrist pin of the Porter-Allen engine. This type which has four guides is used extensively on engines having Tangye frames. The wrist pin has tapered ends.



Figs. 251 and 252.—Murray-Corliss cross head, having cylindrical gibs with wedge adjustment. Besides the adjusting bolts with lock nuts, two other bolts are provided to prevent the possibility of their becoming loose.

Locomotive cross heads are made for one, two and four guides as shown in figs. 253, 254, and 255, 256.

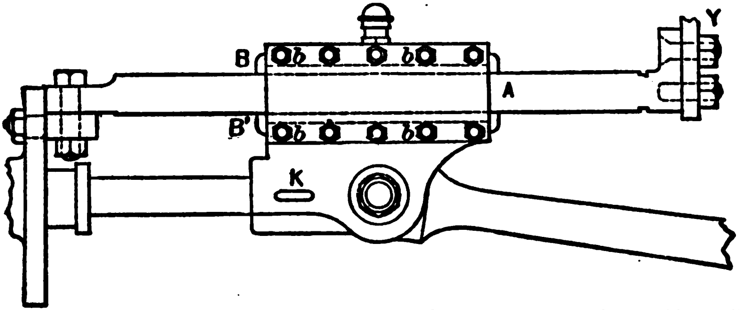


Fig. 253.—Locomotive cross head with single guide. On locomotives the piston rod is usually attached to the cross head by means of a key K. *b*, *b'*, are the gibs, and A, the guide whose outer end is bolted to a transverse piece or *yoke* Y.

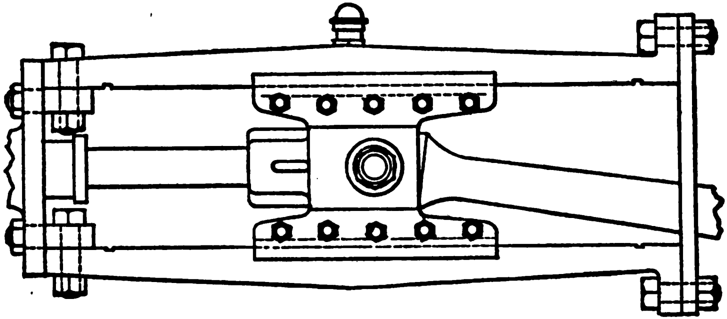
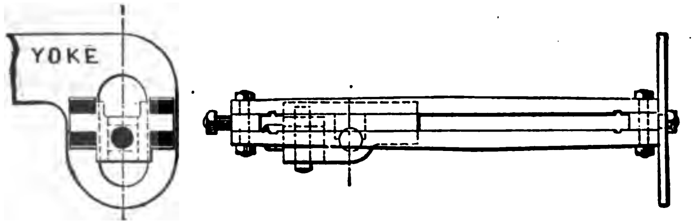


Fig. 254.—Locomotive cross head with double guides

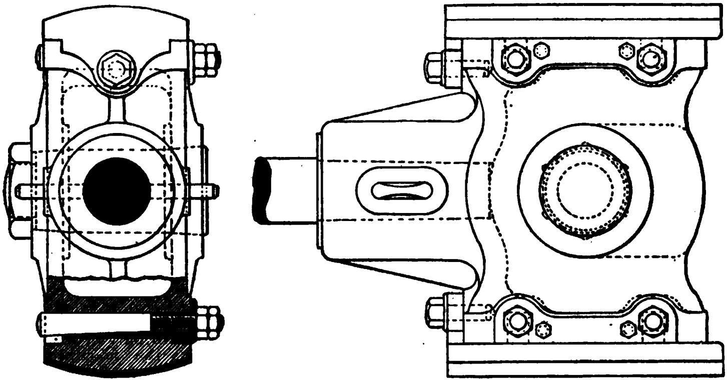


Figs. 255 and 256.—Locomotive cross head with four guides. The end view at the left shows more clearly the location of the guides, and the form of the yoke which supports their outer ends.

In general the rubbing surfaces of the gib and guide may be either plane, inclined or cylindrical.*

The gibs are usually made of some other metal because there is less friction between rubbing surfaces of dissimilar metals than when made of the same metal. Brass, white metal and other alloys are used for gibs. They are usually of brass with babbet, or white metal inserted into grooves or circular holes.

On account of the ease of alignment the cylindrical or turned gib is generally used.



FIGS. 257 and 258.—Fishkill-Corliss crosshead. The cylindrical gibs are adjustable by means of transverse eccentric keys as shown in the sectional end view. A key is used to secure the piston rod.

To allow for wear between the rubbing surfaces, gibs are made adjustable. There are various methods of adjustment, the simplest of which is the insertion of paper liners between the gib and crosshead body.

Another mode of adjustment is by means of inclined surfaces, moved by bolts, or eccentric keys.

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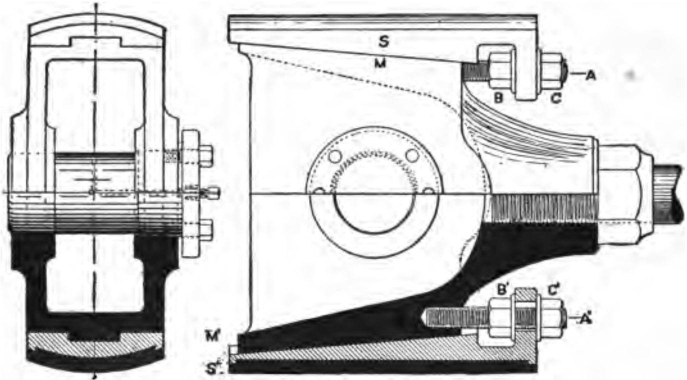
*NOTE.—Cast iron, hard and close grained is considered the best material for guides. Its surface after a few hours' work becomes exceedingly hard and highly polished and offers very little resistance to the gib. So long as this hard skin remains intact, no trouble will be experienced, but if abrasion take place from heating or other cause, it rarely works well afterwards and should at once be planed afresh.



FIGS. 259 and 260.—Fulton-Corliss cross head. The gibs S, S, have cylindrical faces, and are fitted to the inclined surfaces M, M', being adjustable by the nuts B, C, and B', C', on studs A, A'.

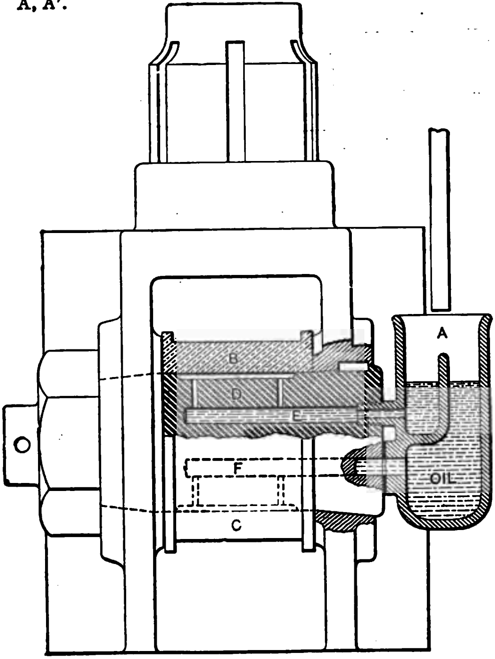


FIG. 261.—Reeves-Cubberley cross head for vertical engine showing sectional view of wrist pin and bearing with oiling device. In the cup there is a partition A, as shown, oil from the vertical pipe fills the oil cup on both sides of the partition A. One side of the partition connects with channel E, through which the top half of the cross head pin D, and connecting rod box B, is lubricated. When pressure is relieved between surface of pin D, and rod bearing B, oil on inside of the partition in cup flushes this space with oil; without this partition A, in the cup, oil would settle to the bottom of cup and the top surface of the pin would not get oil as it now does. Of course, the bottom side of cross head pin and bearing C, is oiled through channel F, in the manner described above.

There are numerous modifications of this mode of adjustment as shown in the accompanying cuts.

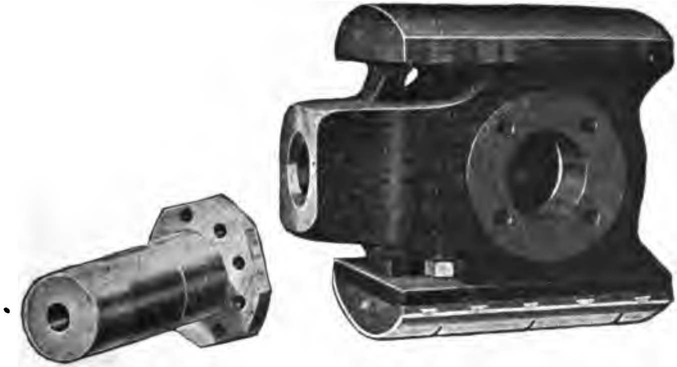


Fig. 262.—Cross head of Ames single valve engine. It is of semi-box section and fitted both on top and bottom with removable shoes spotted with babbitt forming about 40% of the wearing surface. The tapered pin is held in position by four tap bolts passing through the head of the pin. By removing these four bolts and replacing in tapped holes provided in the flange, the pin may easily be drawn from the cross head without the use of a sledge, making it possible to remove and replace the pin from the front side of the engine. The pin may be turned 90° when worn to provide new wearing surfaces.

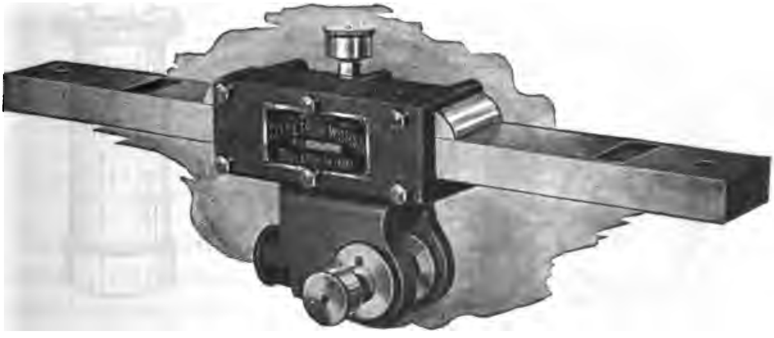
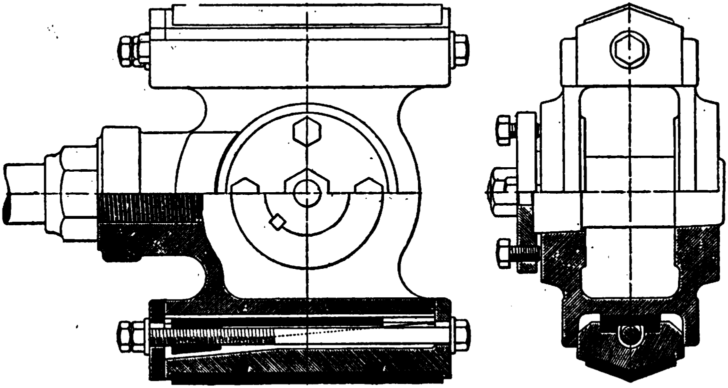
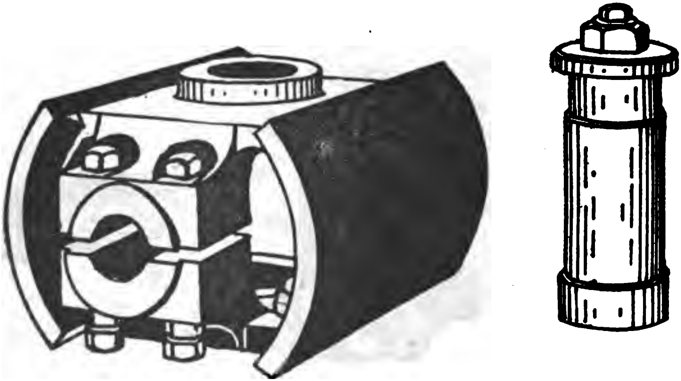


Fig. 263.—Clyde hoisting engine cross head and guide. The cross head is of the single bar locomotive type and is fitted with bronze gibs. In *reversing engines* the cross heads are furnished with bronze gibs for both *top and bottom* sides of guide bar. The bar, as shown, is recessed at each end to permit over-travel of the gib, thus preventing the wearing of shoulders. These recesses also retain the oil.

The first named method of adjustment by inclined surfaces is shown in figs. 259 and 260. The gibs S, S', rest on inclined surfaces M, M', of the cross head, and by moving them to the left they will be spread further apart. By this means wear may be taken up between the rubbing surfaces. A stud A, attached to either side of the cross head, passes through a projection on each gib. The gibs are retained in any position by means of these studs and the nuts B, C, and B', C'.



FIGS. 264 and 265.—Harris-Corliss cross head. The gibs have V shaped faces with wedge adjustment. A movable, concealed wedge operated by the through bolt, permits the adjustment of the gibs without any lengthwise movement of the latter.



FIGS. 266 and 267.—Split type of cross head, and wrist pin. The neck is threaded to receive the piston rod. Instead of the usual lock nut, the neck is split, and the two halves made to grip the rod by means of cross bolts as shown. This type of joint is satisfactory when the machining is carefully done; a loose fit will cause trouble.

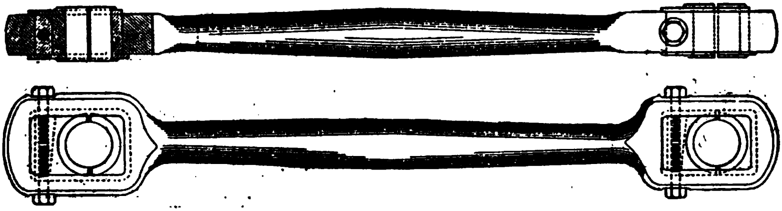
To make the adjustment, nut B, is first loosened and then nut C, tightened until the gib is in the desired position, nut B, is then tightened which locks the gib in place.

There are numerous modifications of this mode of adjustment as shown in the accompanying cuts.

The cross head shown in figs. 264 and 265 has the rubbing surfaces inclined and the adjustment for wear is made by concealed wedges operated by fore and aft adjusting bolts.

Adjustment by means of eccentric keys is shown in figs. 257 and 258. The gibs are fitted with four tapered keys which work crosswise; the construction is plainly shown in the two views.

Cross heads are attached to piston rods by screwed joints, or by means of keys.



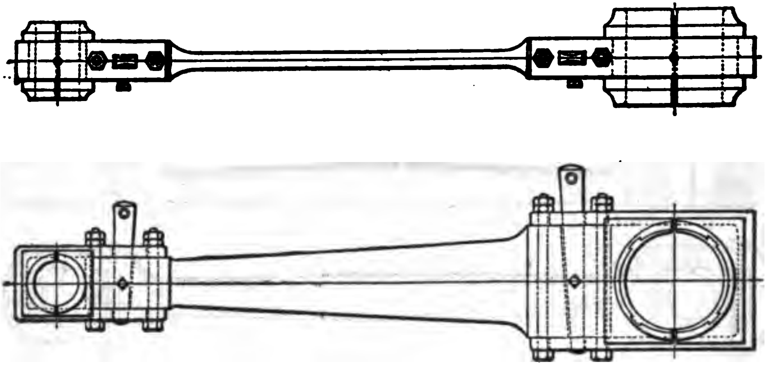
FIGS. 268 and 269.—Eclipse Corliss connecting rod. The type generally used on slow and medium speed engines. The rod of circular section tapers from the middle to the forged ends which contain the crank and wrist pin brasses. *As constructed,* one adjustment lengthens the rod, while the other shortens it, the combined effect is to keep the length the same.

In the first method, either a lock nut is provided to prevent the rod turning, or else the neck of the cross head is split forming a split bushing as shown in fig. 266. The rod, after it is screwed into the bushing, is clamped by two bolts with lock nuts. The principle here employed being the same as with the wrist pin in fig. 239. In either case, first class machine work is necessary as more or less trouble is experienced with a loose fit.

The Connecting Rod.—The to and fro or *reciprocating* motion of the piston is converted into a rotary motion by the connecting rod, which joins the crosshead to the crank. The connection is made by the wrist and crank pins, for which there are suitable bearings at the ends of the rod.

The length of the connecting rod, measured between the centers of the wrist and crank pins, is usually two to two and a half times the length of the stroke; the latter proportion, says Thurston, giving a long and easy working rod, and the former a rather short, but yet a manageable one.

The rod must be strong enough to resist not only the alternate stresses of tension and compression, but also the bending stresses due to its oscillation.



Figs. 270 and 271.—Ball and Wood connecting rod. This is the form of rod used on high speed engines. The rectangular cross section, and the pronounced sidewise taper is the best shape to resist the severe bending strains due to high rotative speed.

For engines of slow and medium rotative speed, the rod is usually of circular cross section, tapered from the center to both ends as shown in figs. 268 and 269.

In high speed engines, the rod is made of rectangular section as shown in figs. 270 and 271, which is a better shape to resist the bending strains.

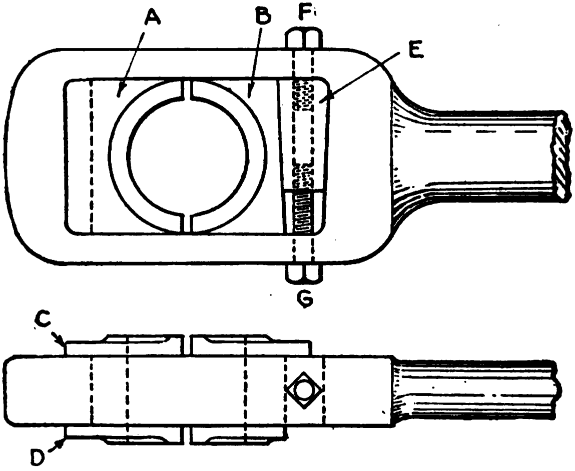
Most connecting rods are made of steel while the bearings or brasses are of brass lined usually with Babbitt metal. The rod ends are solid or built up.

There are various arrangements for adjusting the brasses to take up wear, such as:

1. Blocks;
2. Bolts;
3. Gibs and cotters.

Figs. 272 and 273 show a solid end with block adjustment. A rectangular slot is cut in the enlarged section in which is inserted the brasses A and B, having suitable flanges C and D, to retain them in the slot. A wedge shaped block E is fitted to the slide upon B, thus bringing A and B closer together and reducing the size of the bearing. Two bolts, F and G, are threaded in the ends of the block to secure it in position.

To adjust the bearing, F is first loosened and then G tightened to the desired amount. The block is then locked in place by tightening F.

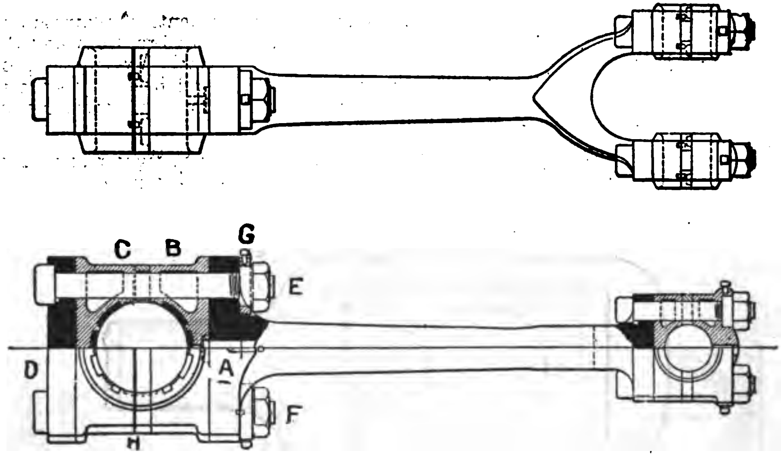


FIGS. 272 and 273.—Solid end with block adjustment; used extensively on Corliss connecting rods for both the wrist pin and crank pin. The parts are: A, B, brasses, A being provided with flanges C, D; E, adjusting block or wedge; F, G, adjustment bolts which retain the block in the desired position.

On marine engines the adjustment is usually made by means of bolts especially at the crank end, as shown in figs. 274 and 275, on account of the ease with which it is disconnected from the crank. The rod ends in a T section A, upon which is placed the brasses B and C and an iron or steel cap D. Two bolts E and F pass through these several members, securing them firmly together. A set screw G locks the bolt nuts.

Sometimes the set screw is omitted and the two nuts provided for each bolt which serves the same purpose. Liners H, are inserted between the brasses to prevent them seizing the pin when the bolts are tightened.

To adjust this bearing the bolts are first loosened and then one or more liners removed or replaced as the case may be, the bolts tightened, and the nuts locked by the set screws.



Figs. 274 and 275.—Marine connecting rod with forked end and double bearing for the wrist pin. To the T end A, is attached the brasses B, C, and cap D, by the bolts E and F.

On large bearings, as shown in fig. 276, the brasses are hollowed out at A, and B, to save metal, some provision being made so that the bolts will not turn with the nuts.*

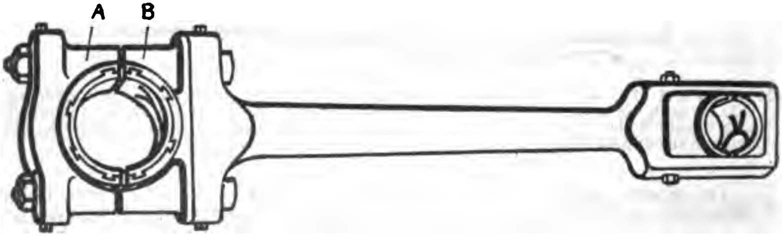


Fig. 276.—The Phoenix connecting rod. The crank pin end is of the marine type while the wrist pin end is solid with block adjustment. The illustration shows the arrangement of oil grooves, and the recesses in the brasses to retain the Babbitt lining in place.

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*NOTE.—The bolts are usually turned with part of their length reduced to the diameter of the bottom of the thread. This makes the bolt more elastic without reducing its strength.

The wrist pin end of a marine rod is made in several ways. It may be:

1. A solid end with block adjustment as shown in fig. 276;

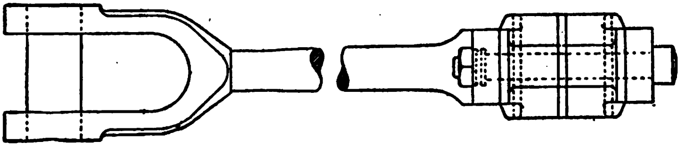


FIG. 277.—Marine connecting rod with forked wrist pin end; the two branches carry the wrist pin.

2. A forked end having two arms to which is attached the wrist pin (fig. 277);
3. A forked end carrying two wrist pin bearings, as shown in figs. 274 and 275.

In the second mentioned construction it should be noted that the wrist pin is made fast to the rod instead of to the crosshead. This requires that the crosshead contain the bearing.

Many marine engines of the smaller sizes have this type of rod end. Figs. 279 to 231 show the style crosshead used with rod in position.

In figs. 232 and 283 is shown a built up rod end with gib and cotter adjustment.* The rod end A which is of enlarged rectangular cross section and the brasses

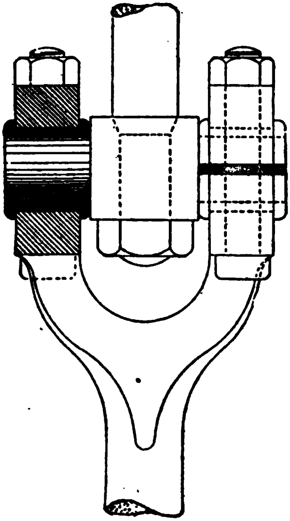
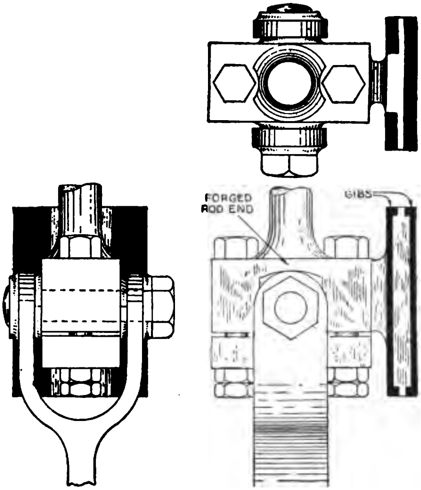


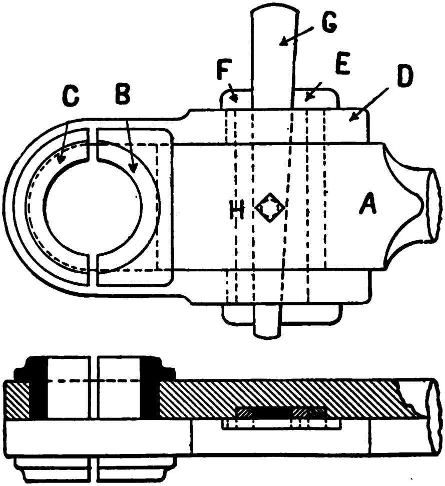
FIG. 278.—Marine connecting rod with forked wrist pin end, containing a double bearing for the wrist pin. The advantage of this form of rod is that the height of the cylinders may be less than with other types; it is difficult, however, to make uniform adjustment of the two bearings.

NOTE—*Continued.*

hence, it is better able to resist the severe shocks that come upon it. To provide for uneven adjustments each bolt is made large enough to carry two-thirds of the load and so proportioned that the stress on it does not exceed 5,000 pounds per square inch at the smallest cross section. Since in adjusting the bolts more pressure is liable to be brought on one than the other, two-thirds of the load should be considered as being carried by one bolt in determining its size.



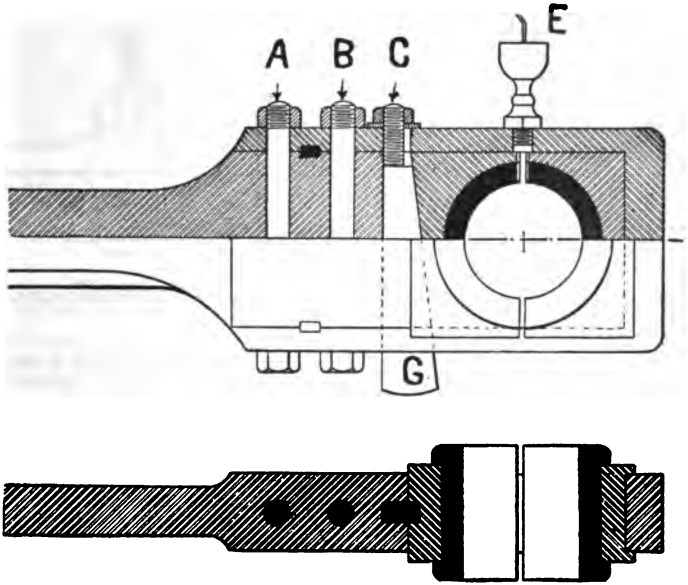
Figs. 279 to 281.—Detail of connecting rod with wrist pin in forked end and cross head containing wrist pin bearing.



Figs. 282 and 283.—Built up connecting rod with gib and cotter adjustment. The parts are: A, stub end of rod; B and C, brasses; D, strap; E and F, cotters; G, gib; H, set screw.

B and C are held together by a strap D, two cotters E and F and the gib G.* The latter is a wedge shaped key, which on being driven in, forces the strap to the right, thus bringing the brasses closer together. The gib is retained in the desired position by the set screw H.

Sometimes the cotters E and F are omitted and the strap fastened to the rod, as shown in figs. 284 and 285, by the bolts A and B. The brasses are adjusted by the gib G which has a threaded end and nut C. At E is shown a wiper oil cup.



FIGS. 284 and 285.—Built up connecting rod cotters. The cross bolts A and B secure the strap to the stub end of the rod. The gib G is adjusted at the stud end C. A wiper oil cup for a horizontal rod is shown at E.

Two other methods of gib adjustment are shown in figs. 286 and 287. In each case the gib is attached to a bolt; in fig. 286 the bolt forms a part of the cotter, and in fig. 287 the cotter is omitted, the bolt being threaded into the rod end.

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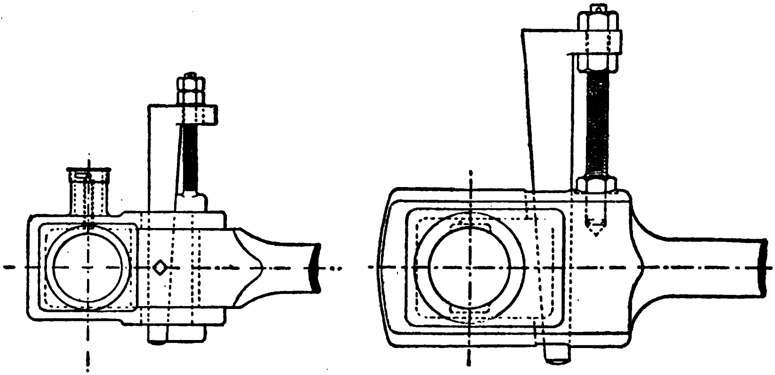
*NOTE.—The thickness of the gib or key is usually one-fourth of the width of the strap, and the breadth parallel to the strap should be such that the cross section will have a shearing strength equal to the tensile strength of the section of the strap. The taper of the gib is generally about five-eighths inch to the foot.

If, as a result of adjusting the brasses of a connecting rod, its length be changed, the clearance at the two ends of the cylinder will be altered.



Figs. 286 and 287.—Two methods of gib adjustment. Fig. 286.—Gib attached to bolt which forms part of the cotter; fig. 287, cotter omitted.

To prevent this, the rod is sometimes constructed in such a way that tightening up the brasses at one end lengthens the rod, and tightening at the other end shortens it as shown in fig. 288, where taking up wear at the crank pin end pushes the outer brass *in* and shortens the rod, while a similar adjustment at the wrist pin end moves the inner brass *outward* and lengthens the rod. Thus, the effect of the two adjustments tends to keep the rod length the same.

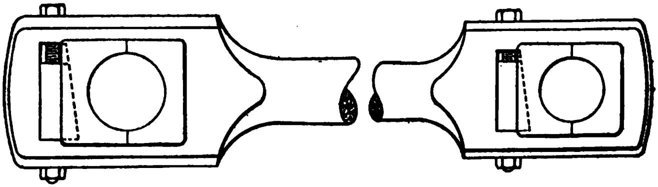


Fig. 288.—Twin City Corliss connecting rod with solid ends and block adjustment. The blocks or adjusting wedges are on the same side of the pins, thus tending to maintain a constant length of rod.

A modification of the solid end rod is shown in fig. 287, which is a desirable rod for a large engine; it is known as the "hatchet end" type.

By this arrangement, when the bolt is removed from its position it allows a side of the strap to be taken out, so that the rod can be easily lifted off the pin. The adjustment of the brasses is made by means of concealed blocks, set up by adjusting bolts.

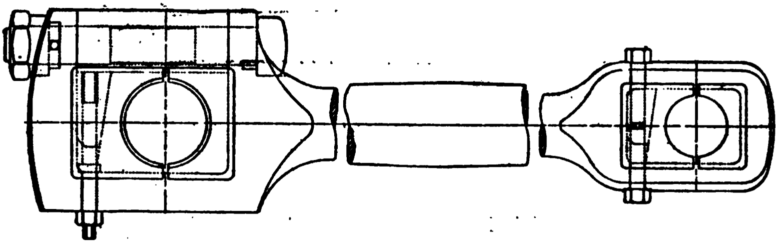


FIG. 289.—The "hatchet" end type of connecting rod as used on the Harris-Corliss engine. A desirable form for large engines, permitting easy removal of the brasses.

On some rods the end is made removable instead of the side. A rod of this kind is shown in fig. 290. The block A at the end is dovetailed to fit the ends of the fork and is held in place by a large bolt as shown. By removing the bolt and block, the bearing may be taken out from the rod for inspection without disturbing the adjustment.

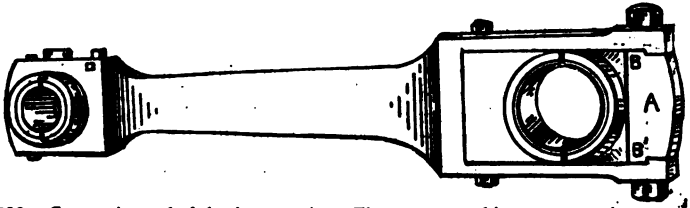


FIG. 290.—Connecting rod of the Ames engine. The extreme end is a separate piece removable by taking out the end bolts. The brasses are then easily accessible. The rod is provided with block adjustment.

In single acting engines it is not considered necessary by some to use refined methods of taking up wear.

Fig. 291 shows the connecting rod used on the larger sizes of Westinghouse vertical single acting engine, no special provision being made for adjustment.

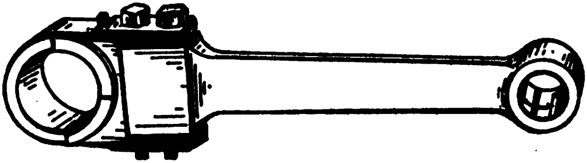


FIG. 291.—The Westinghouse connecting rod for high speed, short stroke engines. On account of the large cylinder diameter in proportion to the stroke, the bearings are of unusual size for the length of rod.

The strap is held rigid in place by two tap bolts passing through the rod end and threaded into one side of the strap. The bronze bearings are lined with Babbitt metal and are kept in place sidewise by flanges which embrace the sides of the strap and rod.

When lost motion becomes unusually large it may be taken up by inserting between the stub end of the rod and the bronze, one or more pieces of thin sheet steel known as *shims* or *liners.*

Fig. 292 illustrates the general proportion of a rod suitable for a short stroke high speed engine.

The rod ends are very large in proportion to the length of the rod. This results from the large cylinder diameter as frequently in engines of this class, the diameter is greater than the length of the cylinder, and the length of the rod with respect to the stroke is usually less than for ordinary service.

Ques. What is a "gudgeon"?

Ans. A gudgeon is an obsolete name for a *wrist pin.*

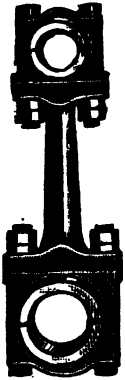


FIG. 292.—Sturtevant connecting rod for vertical single engine. The rod is an open hearth steel forging. The marine type crank pin box is of babbitted malleable iron or semi-steel, dependent on size. The wrist pin box is also of the marine type, except in the larger sizes, which are of the solid end type with wedge adjustment.

Ques. Is the full force exerted on the piston transmitted to the crank?

Ans. Only at the dead centers. At any other point of the stroke, part of the force is transmitted as a side thrust to the guides.

Ques. When a force is thus divided into two or more forces acting in different directions what are they called?

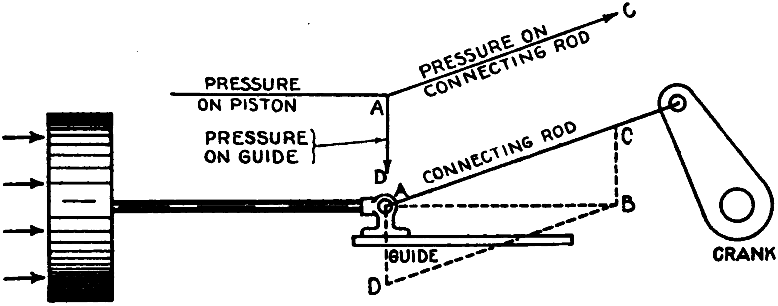


Fig. 293.—Parallelogram of forces showing the two component forces at the crosshead due to the thrust of the piston. By means of this diagram the pressure on the guides and on the crank pin can be obtained.

Ans. *Components.* The original force is called the ***resultant*** because it is the equivalent of the several component forces.

Ques. How may component forces such as those produced by the action of a connecting rod be measured?

Ans. By drawing a *parallelogram of forces* as in fig. 293.

In the skeleton diagram of the moving parts of an engine, the line of the piston is extended, and with any suitable scale the distance A B marked off so that it will represent the total load on the piston.

For instance, if the total load on the piston be 500 pounds, and the scale taken be 100 pounds to the inch, then A B will be five inches.

At A, the center of the wrist pin, the force transmitted to the piston rod by the piston will split up into two component forces, one acting in the direction of the connecting rod and the other acting perpendicular to the guide. The intensity of these forces is found by drawing lines through B, parallel to the directions in which they act, giving the points C and D. By measuring A C and A D, with the same scale as was used in laying off A B, and multiplying by the pounds per inch the intensity of the forces is obtained.

The thrust on the guide, when the connecting rod is at its maximum angle with the axis of the piston rod, may also be found by the formula:

$$\text{Thrust} = p \tan\theta$$

in which

p = total load on the piston,
θ = maximum angle of connecting rod

The angle θ, is the angle whose sine = ½ stroke of piston ÷ length of connecting rod. Its values for rods of different lengths are:

Ratio of length of connecting rod to stroke	2	2½	3
Maximum angle of the connecting rod	14° 29′	11°.33′	9° 36′
Tan θ	.258	.204	.169

Example.—If the total load on the piston be 5,000 lbs., and the length of the rod be 2½ times the stroke, then the maximum thrust on the guide is:

$$5{,}000 \times .204 = 1{,}020 \text{ lbs.}$$

Ques. How does the thrust of the connecting rod act on the crank pin?

Ans. It is split up into two component forces. One acts in the direction of a *tangent** to the circle described by the crank pin *which causes the crank to turn*, and the other acts in the direction of the *axis* of the crank arm *which causes the shaft to press against its bearing.*

Thus, in fig. 294, the thrust of the connecting rod is split up at A, into two component forces, one acting in the direction of the tangent A M, and one in the direction of the axis A O. By laying off A B, equal to the thrust of the connecting rod, and completing the parallelogram of forces, the points C and D, are obtained giving the lines A C and A D, whose lengths represent the *intensity* of these forces.

Ques. What is the component A C, called?

Ans. The *tangential*, or *turning force.*

---

*NOTE.—A tangent to a circle is a straight line drawn through a point on its circumference and perpendicular to a line joining this point at the center.

Ques. What is the nature of this turning force?

Ans. It is always less than the force acting on the piston. It increases from zero at the dead center to a maximum near the center of the stroke and then diminishes to zero at the end of the stroke.

Ques. Since the turning force is always less than the force acting on the piston, is there not a considerable loss of power caused by this peculiar action of the connecting rod?

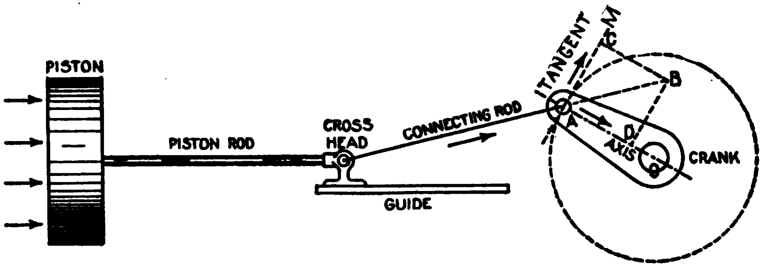


Fig. 294.—Parallelogram of forces showing the two component forces at the crank pin. By means of this diagram the tangential force or "turning effect" can be obtained for any crank position.

Ans. No. Neglecting friction, the same amount of work that is done on the piston is delivered to the crank pin by the connecting rod in turning the shaft.

Ques. · Why is there no loss of power?

Ans. Because during each stroke, the crank pin travels a greater distance than the piston. Hence, the smaller turning force by acting through a longer distance, does the same amount of work as the larger force on the piston acting through the shorter distance.

Work is the product of two factors: *force* and *distance* through which the force acts. These two factors are *inversely proportional* for a given amount of work; that is, if one factor be increased, the other is diminished a like amount. For instance, to raise one pound ten feet requires the same amount of work as is required to raise ten pounds one foot, or two pounds five feet.

Now, in a steam engine, while the crank pin is revolving at a constant *tangential speed**, the speed of the piston is ever varying.

Ques. What is the nature of the motion of the piston?

Ans. It starts from rest at the beginning of the stroke, increasing to a maximum near the middle, then diminishes until it again comes to rest at the end of the stroke.

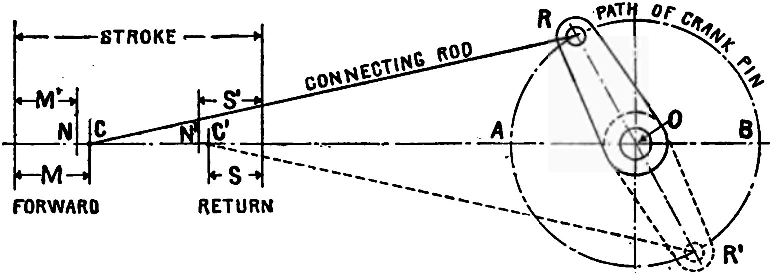


FIG. 295.—Diagram showing the effect of the *angularity of the connecting rod*. For equal crank pin movement from each end of the stroke, the angularity of the rod causes the piston to travel further on the forward stroke, than on the return stroke.

The conditions which prevail at the piston and crank pin clearly illustrate the inverse relations which exist between the factors *force* and *distance* for a given amount of *work* as can be shown by a series of diagrams.

Ques. What effect does the connecting rod have on the movement of the piston?

Ans. Starting at the beginning of the forward stroke, the inclination or *angularity*† of the rod with respect to the cylinder axis, causes the piston to move somewhat more than half its stroke while the crank is moving the first quarter of its revolution, somewhat less than half stroke during the second and third quarters, and again somewhat more than half stroke during the fourth or last quarter of the revolution.

---

*NOTE.—The tangential speed of a point revolving in a circle is its equivalent speed if it were moving in a straight line. Thus, the speed of a belt as it leaves a pulley is the tangential speed of a point on the face of the pulley.

†NOTE.—The "angularity" of the connecting rod is sometimes called the *obliquity*.

In fig. 295 is shown the effect of the angularity of the rod in distorting the movement of the piston. In the diagram the piston is not shown since its position with respect to the stroke corresponds exactly to that of the wrist pin C.

The connecting rod is shown in two positions, C R and C′ R′, such that the crank pin has traveled equal distances A R and B R′ from the dead centers. The piston positions are indicated by C and C′, the piston having traveled on the forward stroke the distance M and on the return stroke the distance S. For equal crank pin travel from each end of the stroke, it is thus seen that the piston travels further on the forward stroke than on the return stroke

Were it not for the angularity of the rod, the piston would travel the equal distance M′ and S′. The connecting rod then increases the piston travel by a distance N C on the forward stroke and diminishes it by a distance N′ C′ on the return stroke.

Ques. On what does the amount of this distortion of the piston movement depend?

Ans. On the length of the connecting rod.

The shorter the rod the greater the distortion.

Ques. What important effect has the angularity of the rod on the steam distribution to the cylinder?

Ans. It causes cut off to occur *too late* on the forward stroke, and *too soon* on the return stroke.

Ques. What is a "Scotch yoke?"

Ans. A device sometimes used instead of a connecting rod.

It consists of a metal frame similar to a Stevenson link but with straight sides as shown in fig. 296. To the center of one side is attached the piston rod. A continuation of the rod from the other side passes through a bearing which prevents any side movement of the yoke. The crank pin passes through a block which slides to and fro in the yoke.

Ques. What are the advantages of a Scotch yoke?

Ans. There is no distortion in the piston movement and the crank shaft may be placed nearer the cylinder than with a connecting rod.

Ques. Why then has it been displaced by the connecting rod?

Ans. Because the good features of a Scotch yoke are more than offset by the friction and wear of the block and by the extended piston rod and outer bearing.

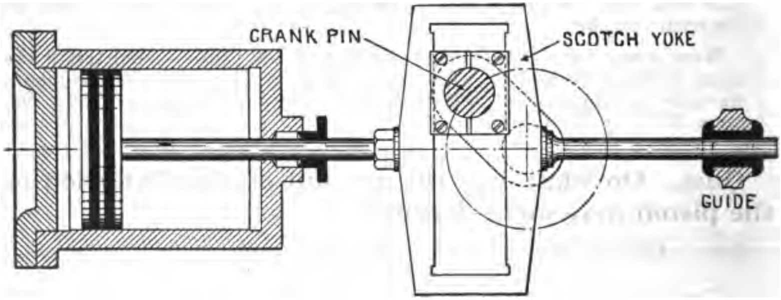


FIG. 296.—The "Scotch yoke." Used to some extent on early engines. Its advantages are overbalanced by several objections, such as friction and wear of block, extended piston rod, outer bearing, etc.

The Crank Shaft.—The to and fro, or reciprocating motion of the piston is converted into rotary motion by the crank shaft which consists of:

1. The shaft;
2. The crank arm;
3. The crank pin.

In construction, the crank shaft is either built up from separate parts or made from a solid forging.

According to the type of engine, it may be classified as:

1. Overhung crank;
2. Center crank;
3. One, two or more throw according to the number of cylinders.

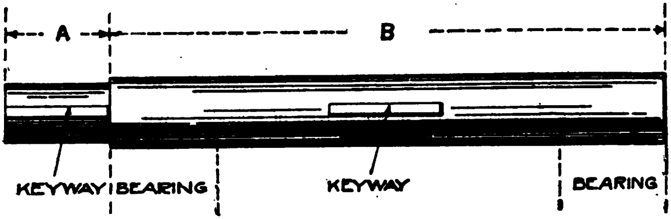


Fig. 297.—Usual form of shaft for a stationary engine. The crank arm is fitted to the end A, which is turned to slightly reduced diameter. The keyway in the central portion is for the fly wheel.

In the built up type as generally used on stationary engines, the shaft itself consists of a cylindrical piece of suitable length as shown in fig. 297.

A portion of the shaft, A, upon which the crank is fastened is sometimes turned to slightly smaller diameter, thus forming a shoulder against which the crank is driven when being fitted. The length B is made such that there is sufficient room for the bearings, valve gear and fly wheel. Two keyways are provided, as shown in the figure, so that the crank and the fly wheel may be keyed to the shaft and thus prevented turning on the latter.

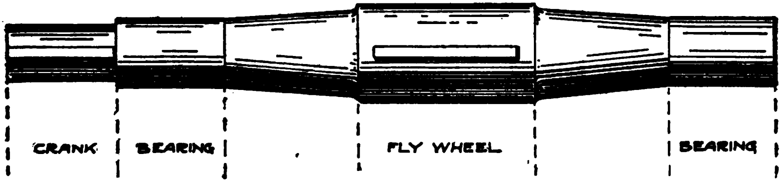


Fig. 298.—Approved form for a long shaft carrying a heavy fly wheel. The enlarged central portion gives stiffness to prevent springing.

On an engine having a heavy fly wheel and long shaft, it is usual to make part of the shaft tapered as shown in fig. 298. This gives great strength to resist bending stresses.

In some cases, instead of reducing the shaft diameter for the crank as in fig. 297, a shoulder is formed by a projection or flange as shown in fig. 299.

The crank consists of an arm with a boss at each end, one to take the main shaft and the other the crank pin. The arm is made solid, or of webbed cross section as shown in figs. 300 and 301.

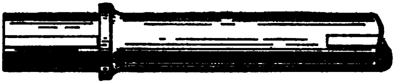
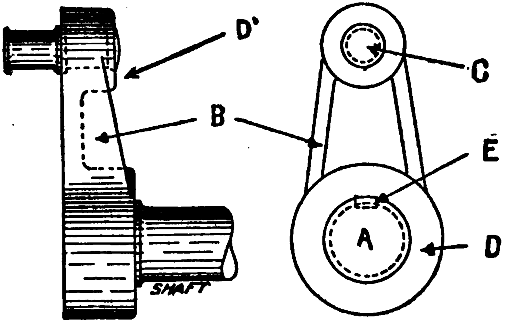


Fig. 299.—Crank end of shaft with flange instead of shoulder. With this construction the diameter of the shaft is not reduced at the crank end.

The crank is secured firmly on the shaft by making it a drive or shrink fit and further secured by a key.*

The shrinking is done by boring out the hole a shade smaller than the shaft, then heating the crank around the hole, thus causing the material to expand and the hole to become larger. The crank is then placed on the shaft, and on cooling it contracts and grips the shaft with great firmness.



Figs. 300 and 301.—Webbed crank arm. The crank is usually fastened to the shaft with a drive, or shrink fit, and further secured by a key. The parts shown are: A, shaft; B, webbed crank arm; C, crank pin; D, boss at shaft end; D′, boss at pin end; E key.

The crank pin is usually forced in place by hydraulic pressure, or fitted by shrinkage and the end riveted as at D′ in fig. 300; as a rule no key is provided.

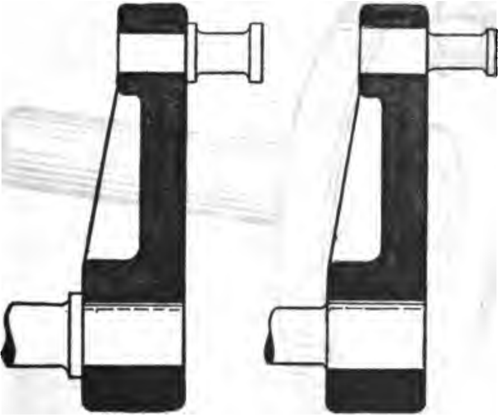
The proportions between the diameter and length of the crank pin vary.

---

*NOTE.—The standard proportions for a key are: width = ¼ of the shaft diameter, thickness = ⅙ of the shaft diameter.

On slow running engines long pins may be used to advantage, but high speed engines require short pins in order to bring the pin closer to the main bearing and thus reduce the bending stresses set up by the inertia of the connecting rod. It should be noted that for a crank pin of given bearing area,* *the longer the pin, the cooler will it run.* This is because the smaller the diameter, the slower the speed of rubbing.

The way in which cranks are shrunk on to crank pins and crank shafts is often objectionable and responsible for many subsequent failures, both of shaft and crank pin.



FIGS. 302 and 303.—Two forms of crank pin. The first is objectionable in that the sharp edge of the hole in the crank arm may cut into the pin and start a crack. This danger is avoided by the construction shown in fig. 303.

Where a shoulder is provided as shown in fig. 302, the result is that when the crank is shrunk into position on the pin or shaft, as the case may be, the sharp edge of the hole in the crank cuts into the material with a shearing action and starts a crack which afterwards, under the influence of alternating stresses, develops into a fracture, and frequently, as experience has shown, leads to a serious breakdown. To avoid this, the crank and shaft may be constructed as shown in fig. 303, the part which is shrunk on the crank being of slightly larger diameter and of a length exactly equal to the thickness of the crank, so that the shrinkage of the crank has no tendency to cut into the material and so start an incipient fracture.

---

*NOTE.—The size of the crank pin should be such that the pressure per square inch of projected bearing area (that is, the diameter multiplied by the length) should not exceed 300 to 400 pounds for stationary engines, 400 to 500 pounds for marine engines and 800 to 900 pounds for paddle wheel engines.

The radius of the crank arm is measured from the center of the shaft to the center of the crank pin. The *throw* of the crank is equal to the diameter of the crank pin path, that is, the stroke of the piston.

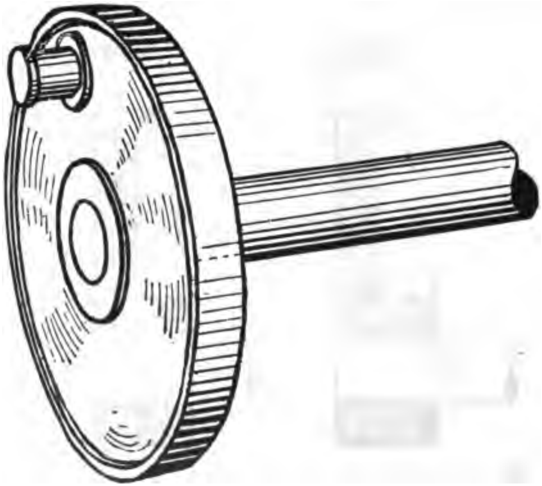


Fig. 304.—Built up crank shaft of the Rollins engine. The crank pin is carried by a disc which is counter-balanced. Hydraulic pressure is used to force the pin and shaft in place.

Built up crank shafts are sometimes constructed with a cast iron disc instead of a crank arm as shown in fig. 304.

The portion of the disc opposite the pin is usually made thicker than the other part, the extra weight being used as a balance to the weights of the reciprocating parts and known as a *counter-weight*. In high speed engines this is desirable to reduce vibration. The disc is attached to the shaft by the methods just described.

The center crank shaft is usually composed of two lengths of shaft, each provided with a disc or arm and having one crank pin joining the two, as shown in fig. 305.

This is the construction usually found in stationary engines.

In the higher class of engine construction, the crank shaft is made from a solid forging for the small and medium sizes, the rough forging as delivered from the forge hammer has the form as shown in fig. 306. The crank is first slotted out (fig. 307), then turned in the lathe and finished as shown in fig. 308. When counter-weights are provided, they are usually made separate and clamped to the crank arms.

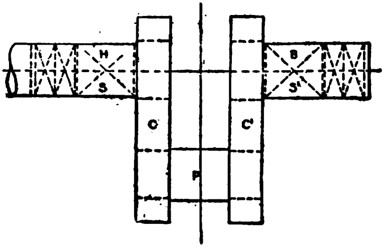
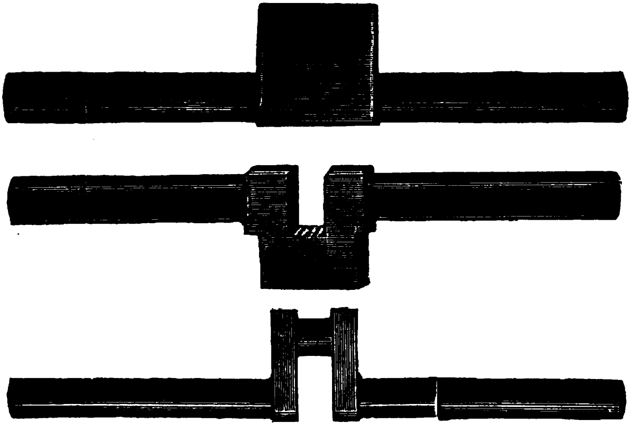


Fig. 305.—Built up crank shaft, consisting of two shaft lengths S, S′; two cranks C, C′, and a crank pin P. In the built up crank shaft the pieces are usually fitted together by shrinking and keying.



Figs. 306 to 308.—Construction of a forged center crank shaft. Fig. 306 shows the rough forging. It is first placed in a slotting machine which removes the metal between the crank arms, as in fig. 307, and then turned in a lathe and finished as shown in fig. 308.

Fig. 309 shows a forged crank shaft with counter-weights attached by transverse bolts. There are various methods of attaching these weights. In figs. 310 and 312, the discs have V shaped grooves in the side

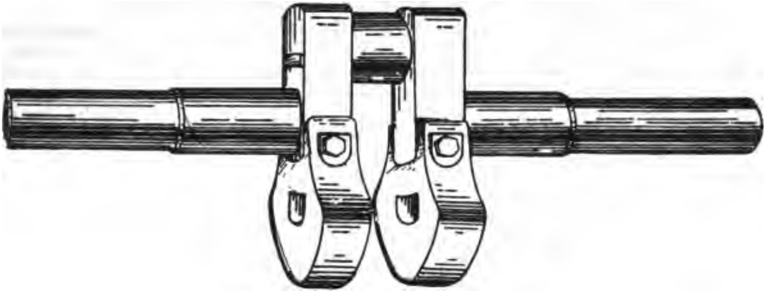
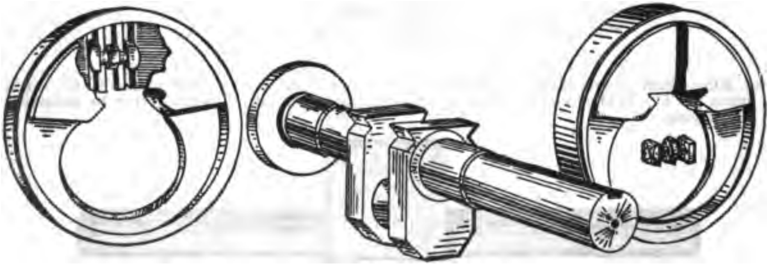


Fig. 309.—Crank shaft of the Erieco engine having counterweights attached by transverse bolts.



Figs. 310 and 312.—Crank shaft of the Watertown engine. The counter balance discs are fitted to V shaped grooves.

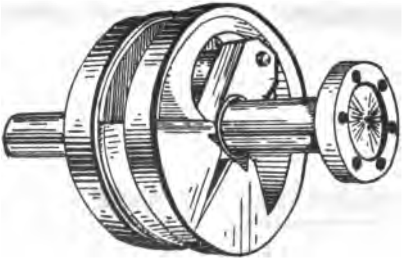
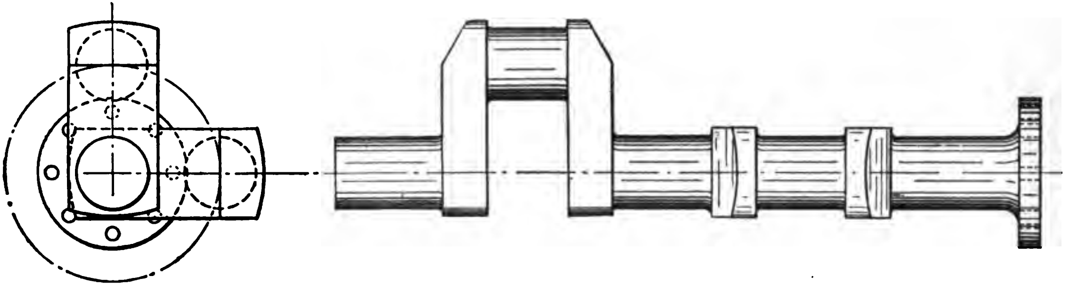
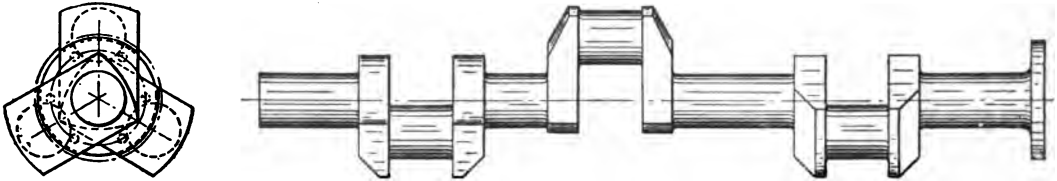


Fig. 313.—Crank shaft assembly of Watertown engine, shown disassembled in figs. 310 to 312.



FIGS. 314 and 315.—Two throw crank shaft for a compound engine. The two cranks are usually set at 90° as illustrated.



FIGS. 316 and 317.—Three throw crank shaft for a triple expansion engine having cranks at 120°. This sequence of cranks produces an excellent "turning effect," giving six impulses per revolution.

of the crank and clamped in place by a bolt with right and left hand threads. The nuts on the end of the bolt are held in grooved recesses in the body of the disc, the latter being cast hollow and loaded with lead counterweights.

In multi-cylinder engines the crank shaft with its two or more cranks is usually a solid forging for the small and medium sizes, and built up for large engines.

Figs. 314 and 315 show a two throw crank shaft for a compound engine, and figs. 316 and 317 a three throw for a triple expansion engine. The former has its cranks set at 90° and the latter at 120°. The order of the crank positions is called the *sequence of cranks.*

Crank shafts for marine engines, when about 10 inches diameter are generally made in duplicate halves, so that in case of damage to a part only half the shaft is condemned, and a spare half shaft can be carried on foreign voyages.

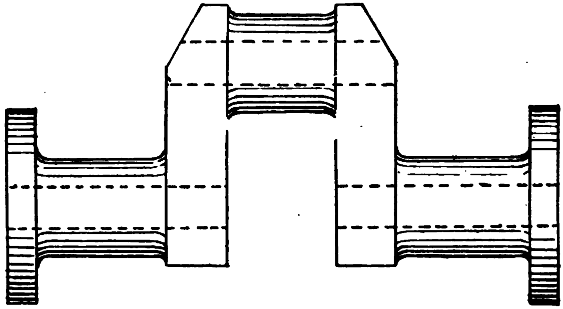


Fig. 318.—Duplicate section for large multi-cylinder crank shaft.

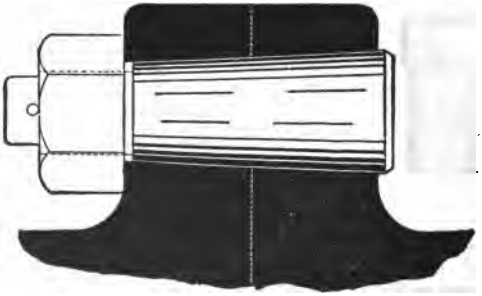


Fig. 319.—Taper flange bolts for connecting sections of large built up crank shafts. The absence of a hexagonal head improves the appearance of the joint.

By this plan there is less labor replacing the damaged half than if the whole shaft be moved.

Flange couplings are usually provided at both ends, so as to be reversible in case of a flaw showing near the after end. Crank arms are forged with the shaft ends, or shrunk on and keyed. The pins are usually shrunk into eyes in the arms.

Large crank shafts are usually built up from duplicate sections, there being a section for each crank. These have flange connections as shown in fig. 318. The several sections being connected by ordinary bolts or taper bolts as shown in fig. 319. The latter type of bolt requires no head, thus giving the flange connections a less clumsy appearance.

The Main Bearings.—These are the bearings in which the crank shaft turns as distinguished from other bearings of the engine. The object of the main bearings is to support the weight of the shaft and fly wheel, and to hold the former in place at right angles to the axis of the cylinder, also to receive the pressure due to that portion or component of the thrust in the connecting rod which is not spent in turning the crank.*

The three requirements of a bearing are:

1. That it be of such size that the pressure of the shaft on each square inch of the bearing will be sufficiently low to prevent heating;
2. That there be means of conveying oil to the rubbing surfaces;
3. Provision for adjusting the bearing to take up wear.

A simple bearing consists of a box, cap, two brasses, liners and bolts to hold the parts together. This kind of bearing is used on vertical engines as shown in fig. 320.

The upper bearing surface or brass B is let into lower brass B′, B being fitted into a cap C. Both brasses are kept from turning by dowel pins D and D′. The brasses are cut away at L and L′ and the space filled with thin strips of sheet metal or liners; these, together, with the brasses are held firmly in place by the bolts M and M′.

---

*NOTE.—This force was explained in the section on connecting rods.

A small hole O, is drilled through the cap and upper brass to convey oil from the lubricating device to the bearing and shaft.

Adjustment for wear is made by taking off the upper bearing and removing one or more liners from each side. The cap is then replaced and

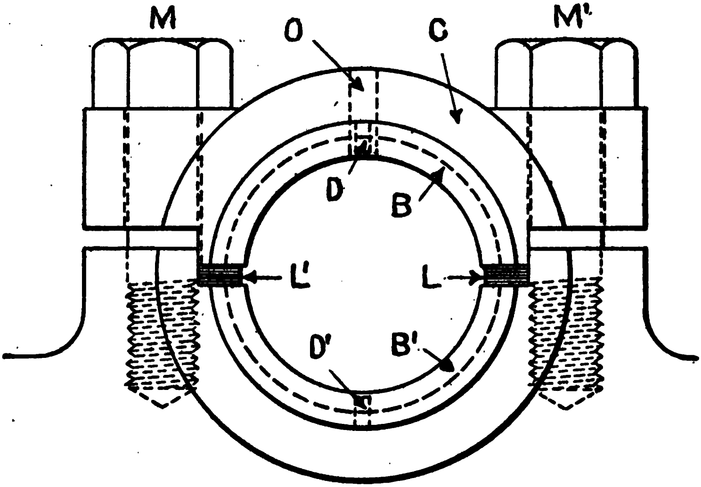


FIG. 320.—Main bearing for a vertical engine; adjustable by means of the bolts and liners on the sides.

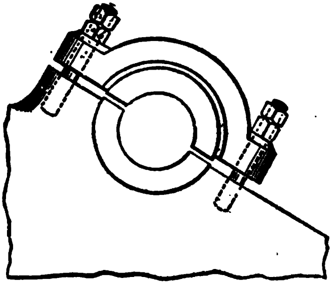


FIG. 321.—Main bearing with liner adjustment for a horizontal engine. The brasses are divided obliquely so that the resultant thrust of the shaft will come centrally, and not at the joints.

the parts again firmly bolted together. Some of the liners are very thin so that adjustment may be made with precision.

This form of bearing is sometimes used on horizontal engines, in which case the brasses are divided obliquely instead of horizontally, so that the resultant thrust of the shaft will press against the brasses centrally and not in the direction of the liners. This construction is shown in fig. 321.

The type of bearing just described is called a *two piece bearing* in distinction from the more complicated form, or *four piece bearing* as generally used on horizontal engines.

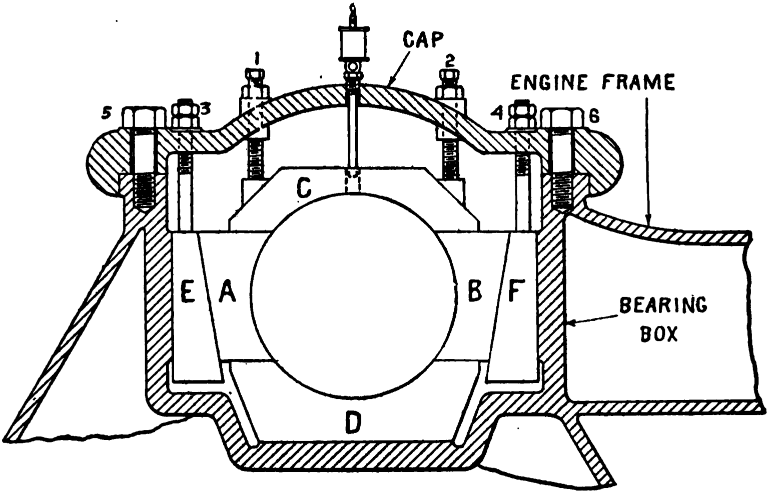


Fig. 322.—A "four piece" main bearing, as generally used on Corliss and other horizontal engines. There are two side brasses A and B, an upper brass C, and a lower brass D. Owing to the great weight of the wheel, little or no pressure comes on the upper brass C. The greatest wear comes on the side brasses, which are adjusted by means of the wedges E', and F.

In this bearing the brasses are divided into four parts, because, on medium and low (rotative) speed horizontal engines, having large and heavy fly wheels, the resulting pressure of the shaft on the bearing is practically in a horizontal direction. Hence, in order to have this pressure come centrally on a brass instead of at the junction of two brasses, they are arranged as shown in fig. 322.

A and B, are two side brasses, and C and D, upper and lower brasses. C and D, are made long enough so that at their extremities, they rest upon A and B; hence, by means of the two adjustable bolts 1 and 2, the four brasses are prevented moving in a vertical direction, the cap being secured by the bolts 5 and 6. The side brasses A and B, have their outer sides inclined which abut against the adjustable wedges E and F.

To take up wear, the side brasses are forced nearer together by adjusting the bolts 3 and 4, which are attached to the wedges E and F. The upper brass C, may be adjusted by filing off sufficient metal at the lower extremities, and tightening bolts 1 and 2.

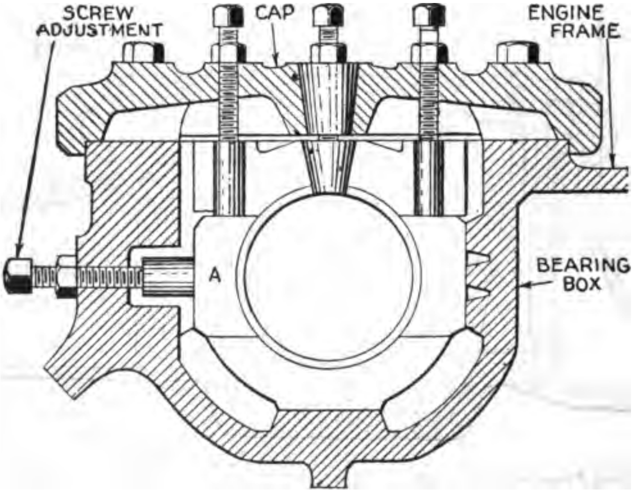
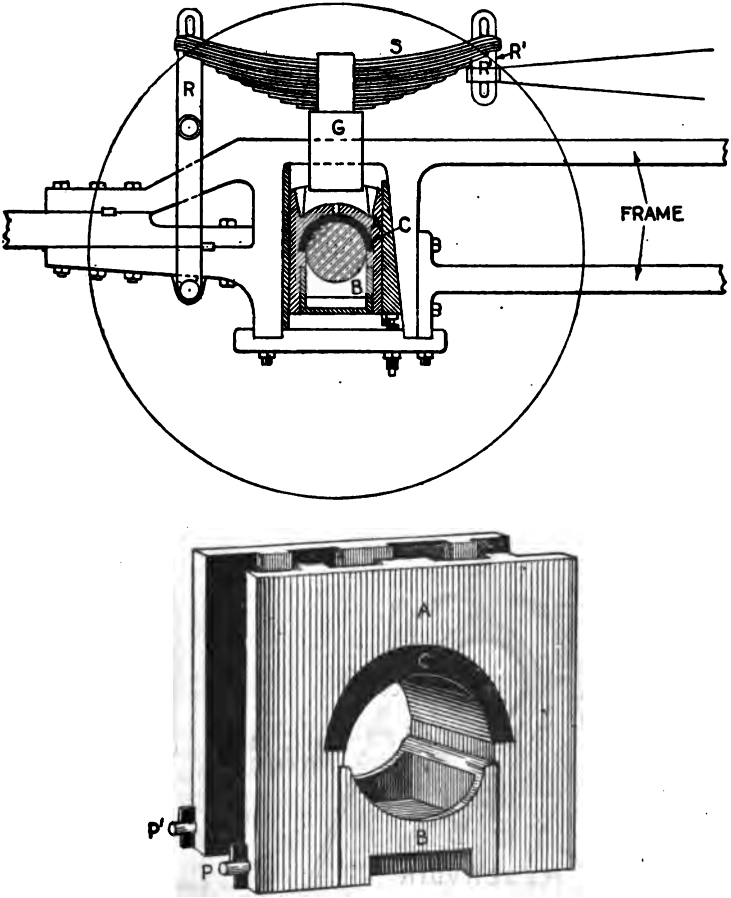


Fig. 323.—Main bearing of the Skinner engine having screw adjustment, of the outer brass.

Sometimes liners are inserted between the upper and side brasses, forming an easy mode of adjustment.

The lower brass D, is raised for wear by inserting liners between it and the bearing box. In some designs wedge adjustment is provided for the lower brass.

The side brasses are sometimes fitted with screw adjustment in place of wedges. On some engines, as shown in fig. 323, only one outer side brass A, is adjustable. In this construction, the adjustment always being made on one side changes the position of the shaft which makes the cylinder clearance unequal, unless liners be inserted between box and opposite brass.



Figs. 324 and 325.—Detail of locomotive driving journal box and assembly in frame. In fig. 324, A, is the box proper which carries part of the weight of the engine, C, being the bearing. Underneath a receptacle B, is filled with cotton waste which is saturated with oil for lubrication. B, is held in position by two pins P P′, as in fig. 325. The box is arranged so as to slide up and down in the jaws of the frame. A spring S, is then placed over the box and above the frame as shown, resting on a W, shaped saddle G, which bears on the top of the box. The frame is suspended to the end of the spring by rods or bars R R′, called spring hangers. Since the boiler and most of the other parts are fastened to the frames, their weight is suspended on the ends of the springs, which cushion the weight they bear.

Figs. 324 and 325 show detail of locomotive journal box and assembly in frame.

On stationary engines, not self-contained, the second bearing for the shaft is called the *outboard bearing*, and as the stresses

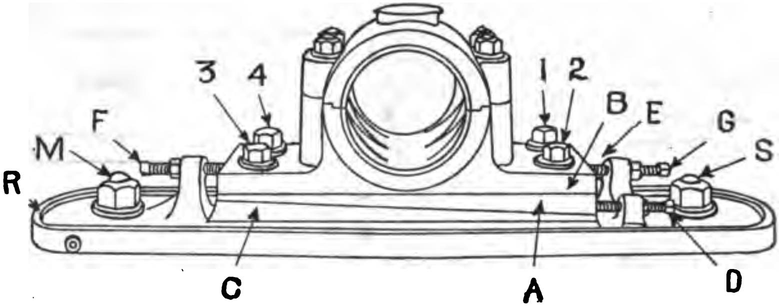
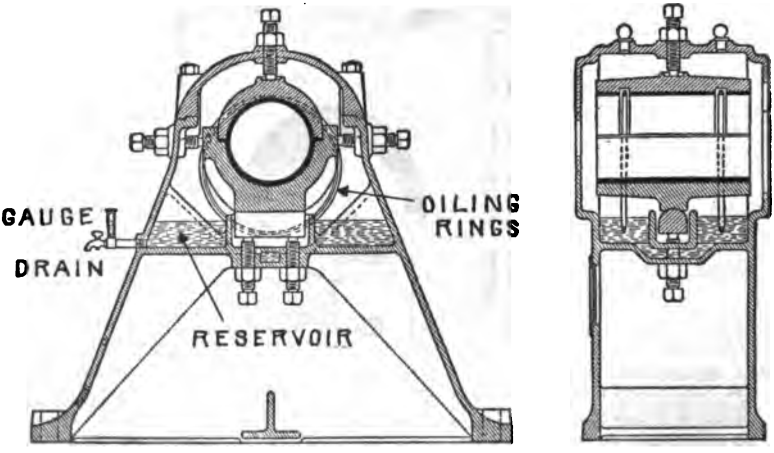


FIG. 326.—Outboard bearing and pillow block of the Murray-Corliss engine. By means of the wedge, bolts, and set screws as shown, the position of the bearing may be adjusted either vertically or horizontally. All engines that are not self-contained should have this type of outboard bearing to secure precision in alignment.

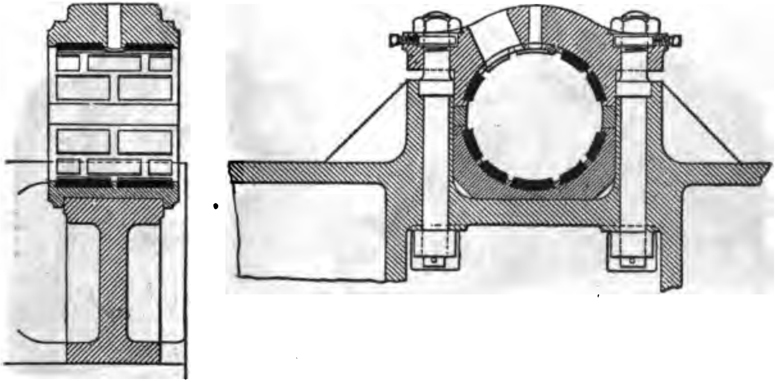


FIGS. 327 and 328.—Self-oiling bearing. The oiling rings which dip into the oil reservoir beneath the bearing, in turning with the shaft, carries oil up to the bearing. A glass gauge at the left indicates the height of the oil in the reservoir.

here are less severe, simpler means of adjustment for the brasses are provided.

Other important adjustments are here necessary. In erecting the engine, it would be difficult to get the bearing in line if it were attached direct to the foundation, hence provision is made whereby the bearing may be moved both up and down, and sidewise. The bearing together with this means of adjustment is called a *pillow block*, the usual construction being shown in fig. 326. A wedge A, is inserted between the base B, of the bearing and the base plate C. By turning the screws D, and E, the wedge is moved along the inclined surface F which raises or lowers the bearing.

A sidewise adjustment is made by means of the screws F and G.



Figs, 329 and 330.—Detail of main bearing of a marine engine showing method of fastening the bearing bolts on large engines.

In making these adjustments the holding down bolts 1, 2, 3, 4 are first loosened and then tightened after making the adjustment. Two large anchor bolts M, S, secure the pillow block to the foundation.

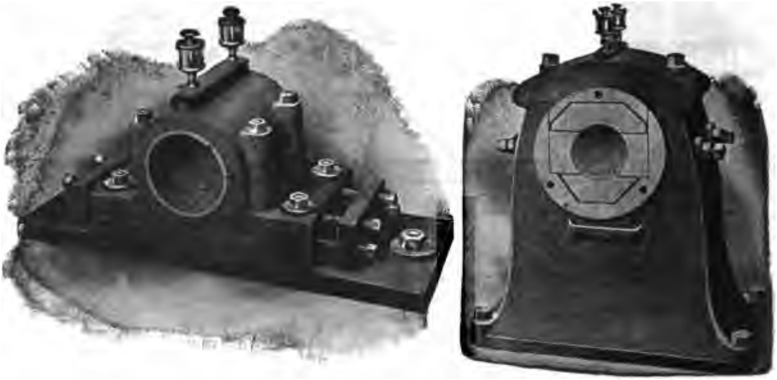
A projecting rim R extends around the base plate which retains the waste oil from the bearing. Usually a pipe is attached to the base plate to allow the oil to drain a vessel.

The Fly Wheel.—In order to keep the reciprocating parts of a steam engine in motion at the dead centers, a large heavy wheel is attached to the shaft which by its *momentum** acts as a reservoir of energy.

***NOTE.—Momentum is the power of overcoming resistance possessed by a body by reason of its motion and weight. It is that which makes a moving body hard to stop.**

In other words, the excess power produced by the engine in the early part of the stroke is stored up in the fly wheel, and given out by it in the latter part where little or no power is developed on account of the expansion of the steam and the engine passing the dead center.

The fly wheel, therefore, on account of its inertia†, tends to keep the speed constant in spite of the variable turning effect produced during the stroke.



Figs. 331 and 332.—Southern pillow block and pedestal. The pillow block has both vertical and horizontal adjustments, whereby the engine shaft may be readily adjusted without jacking up shaft. The construction is such that the pillow block may be removed from sole plate without disconnecting the latter from foundation. The journal is lined with Babbitt metal. The pedestal bearing was designed to meet the demand for an outer bearing that would add to the appearance of engine and engine room, and avoid the necessity of extending masonry of outer pier through floor of engine room, which is always more or less objectionable. The base has large bearing surface, and rests in same plane as engine. Wear is taken up by means of adjusting screws and quarter boxes; the upper and lower bearings adjust themselves automatically to the shaft. Pedestal has oil catch basins, and approved method of lubrication.

---

†NOTE.—***Inertia*** is that property of a body on account of which it tends to continue in the state of rest or motion in which it may be placed, until acted upon by some force.

NOTE.—***The moment of inertia*** of the weight of a body with respect to an axis is ***the algebraic sum of the products of the weight of each elementary particle by the square of its distance from the axis.*** The moment of inertia varies in the same body, according to the position of the axis. It is the least possible when the axis passes through the center of gravity. ***To find the moment of inertia in a body,*** referred to a given axis, ***divide the body into small parts of regular figure. Multiply the weight of each part by the square of the distance of its center of gravity from the axis. The sum of the product is the moment of inertia.***

In the case of a single crank engine the variation of the turning effect is large, while with triple expansion engines having three cranks at 120°, the variation is reduced considerably. It is clear, therefore, that a large and heavy fly wheel is more necessary for a single crank engine than for one of the same power with three cranks.

The four cycle gas engine is an example of extreme fly wheel requirements. Here there is only one impulse or power stroke in two revolutions; hence, the fly wheel must receive, during the power stroke, enough energy to keep the engine moving at approximately uniform speed during the three non-power strokes against the back pressure of exhaust, suction and compression. The large fly wheels fitted to gas engines clearly indicate the variable and intermittent nature of the turning effect.*

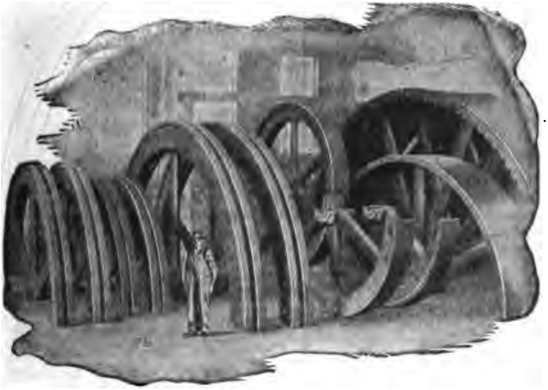


FIG. 333.—Various types of Vilter fly wheels.

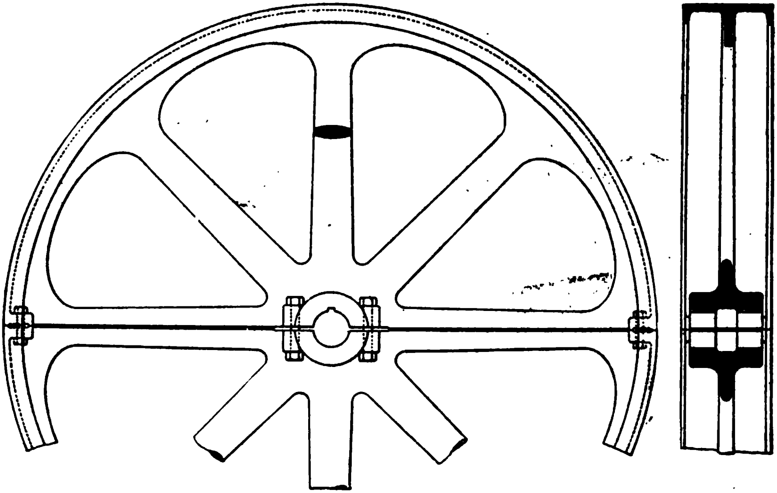
On small engines, power is usually transmitted direct from the fly wheel, there being no separate pulley for the belt.

The diameter of the fly wheel is governed by conditions of service, and limited by the *tangential velocity*, that is, by the speed at the rim, or its equivalent, the belt speed.

---

*NOTE.—So far as turning effect, that is, the number of impulses per revolution, is concerned, one double acting steam cylinder is equivalent to two gas engine cylinders of the two cycle type, or four, of the four cycle type.

Owing to the *centrifugal force** which increases with the speed and which tends to burst the wheel, it is not advisable to run fly wheels at a rim speed higher than 6,000 feet per minute, or roughly, a mile a minute.†



Figs. 334 and 335.—Harris-Corliss ordinary split belt wheel. Wheels of 9 feet and less diameter are made whole, and those from 10 ft. to 17 ft., are made in halves fastened by turned bolts driven into reamed holes. Each half is provided with four oval arms, a center rib, increasing in depth toward the arm, and a return flange follows the outer edge of the wheel on both sides. All wheels above 14 ft. diameter, made in halves, have the rim joints made through the central arms, instead of between arms.

Example.—How large a fly wheel could be safely used on an engine making 200 revolutions per minute?

---

*NOTE.—This is the force which acts on a body revolving in a circular path, tending to force it farther from the center of the circle, because all moving bodies move in straight lines when not acted upon by external forces.

†NOTE.—For any given material, as cast iron, the strength to resist centrifugal force depends only on the velocity of the rim, and not upon its bulk or weight. Chas. T. Porter states that no case of the bursting of a fly wheel with a solid rim in a high speed engine is known. He attributes the bursting of wheels built in segments to insufficient strength of the flanges and bolts by which the segments are held together.

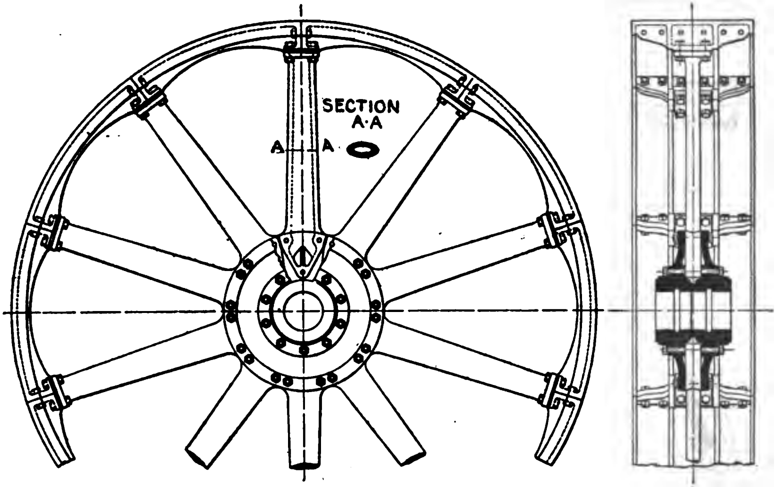
Taking 6,000 feet per minute as the limit of rim speed, the distance traveled per revolution by a point on the rim, or the circumference of the wheel is:

$$6{,}000 \div 200 = 30 \text{ feet.}$$

from which, the diameter corresponding is:

$$30 \div \pi^{*} = 9.5 \text{ feet (approximately).}$$

It is important that an engineer should be able to determine the size of belt required to transmit a given horse power. There are any number of rules for belt sizes, and the results obtained by their application are quite varied. The following rule which is easily remembered, will be found suitable for all ordinary cases;



FIGS. 336 and 337.—Harris-Corliss segmental belt wheel as used on large engines. Wheels 18 feet and upward in diameter are constructed in segments, having 8, 10 or 12 segments in each wheel, and the same number of arms. The latter are of oval hollow construction, as shown in the section A A, this being the form which gives maximum strength. The flanges are planed to a fit. The arms are bolted to the rim segments and are held at the shaft between the hub flanges.

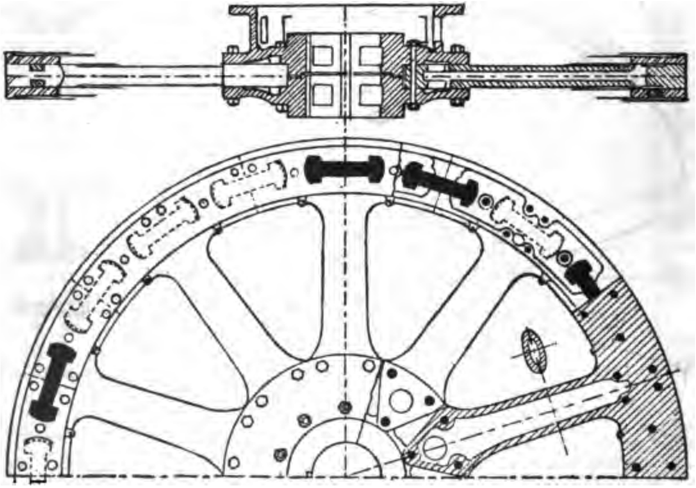
---

*NOTE.—π (pronounced pi) is a greek letter used to denote the ratio between the diameter and circumference of a circle. Its value is 3.1416 (nearly), that is, the circumference of a circle is equal to 3.1416 multiplied by the diameter.

it gives a belt width amply large to deliver the power without undue strain or wear.

Rule.—*A single belt one inch wide, traveling 1,000 feet per minute will transmit one indicated horse power.* A double belt will transmit twice this amount.*

Example.—What size double belt would be required for a 11" × 24" Corliss Engine with an 8 foot †band fly wheel, running at 110 revolutions per minute and developing 60 indicated horse power?



Figs. 338 and 339.—Large fly wheel for Corliss engine. In this wheel, which is 28 ft. in diameter, the rim is made in segments and joined by heavy I links. The hub is in halves, and between the flanges of these halves, the inner ends of the arms are bolted. The unit division of the body of the wheel is one arm with its segment. The rim center is reinforced by side plates of cast steel, each of which covers the angle of two arm spaces, and they break joint on the two sides so that there is nowhere more than one link joint at any cross section of the rim. The whole rim is strongly fastened together by stout pins, which are forced into reamed holes by hydraulic pressure, then riveted cold.

---

*NOTE.—This corresponds to a working strain of 33 pounds per inch of width. Some authorities give for single belts in good condition a working tension of 45 pounds per inch of width, and 64 pounds for a double belt.

†NOTE.—The term "band fly wheel" means that the wheel has a wide rim for the belt so that no pulley is required. When a pulley is used, the rim is made narrow and thick so that the wheel will not take up much room in the direction of the shaft.

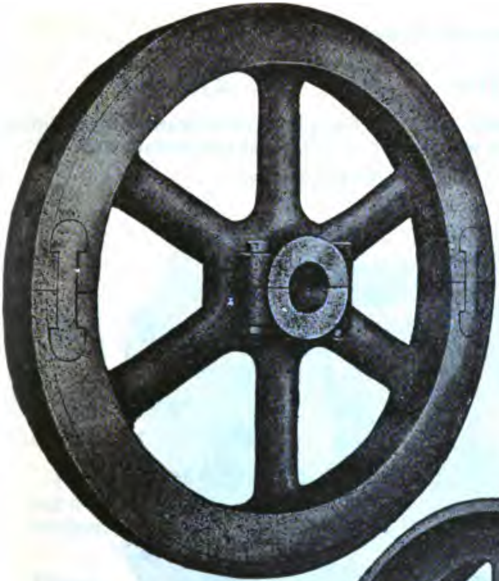


FIG. 340.—Providence balance fly wheel made in halves, principally used on engines directly connected to electric generators, to a line of shafting, or in any service where belts or ropes are not used to transmit the power from the engine.

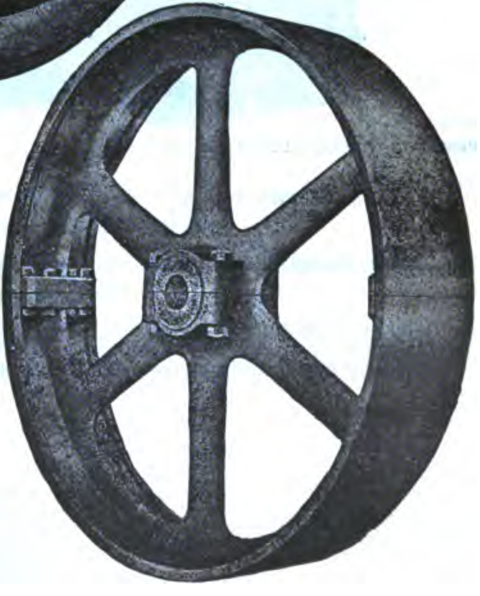


FIG. 341.—Providence pulley wheel made in halves. The rim joint is made at a point midway between the arms as shown.

The rim speed per minute of the fly wheel is:

$$\left.\begin{array}{ccc}\text{Diameter}\times & \pi & \times\ \text{rev. per min.}\\ 8 & \times 3.1416\times & 110\end{array}\right\} = \left\{\begin{array}{l}\text{ft. per min.}\\ 2{,}765\ \text{feet.}\end{array}\right.$$

Since a single belt, traveling at 1,000 feet per minute, transmits one horse power per inch of width it will when run 2,765 feet per minute transmit:

$$2{,}765 \div 1{,}000 = 2.765 \text{ horse power.}$$

FIGS. 342.—Turning a large Vilter rope wheel.

A double belt will transmit twice this amount or

$$2.765 \times 2 = 5.53 \text{ horse power.}$$

Hence, the width of a double belt for 60 horse power is

$$60 \div 5.53 = 11 \text{ inches.}$$

The fly wheel rim should be somewhat wider than the belt, so as to leave a margin of ¼ to ½ inch on each side. This prevents the belt over running the rim by any sidewise movement due to uneven working.

CHAPTER 4

THE SLIDE VALVE

Ques. What is a slide valve?*

Ans. A slide valve is a long rectangular boxlike casting designed to secure the proper distribution of steam to and from the cylinder.

Its general form is shown in fig. 343. Here a portion of the valve is cut away exposing to view the exhaust edge of the valve, the exhaust port, bridges, and one of the steam ports.

Ques. What other name is given to the slide valve?

Ans. It is sometimes called the *simple D valve*†, on account of its resemblance to the capital letter D turned with the flat side down, and having that side practically all removed, as shown in the black cutaway section in fig. 343.

Ques. What are the requirements of a slide valve with respect to the distribution of the steam?

Ans. Considering first only one end of the cylinder, it must: 1, admit steam to the cylinder *just before* the beginning of the forward stroke; 2, *cut off* the supply of steam at some predetermined point of the stroke thus permitting expansion; 3, *release* the steam just before the end of the stroke, allowing it to exhaust the greater part of the return stroke, and 4, close the exhaust at some point before steam is again admitted, in order to save a portion of the exhaust steam, and to *compress* it, thus increasing the back pressure which helps to bring the reciprocating parts to a state of rest in approaching the dead center.

---

*NOTE.—In its broad sense the term "slide valve" includes all sliding valves, as distinguished from rotary valves.

†NOTE.—The slide valve in its crude form was invented by Matthew Murray of Leeds, England, toward the end of the eighteenth century. It was improved upon by James Watt, but the simple D slide valve in use today, is credited to Murdock, an assistant of Watt. It came into general use with the introduction of the locomotive, although Oliver Eames, of Philadelphia, appears to have realized its value, in fact, for years before the advent of the locomotive he applied it to engines of his own make.

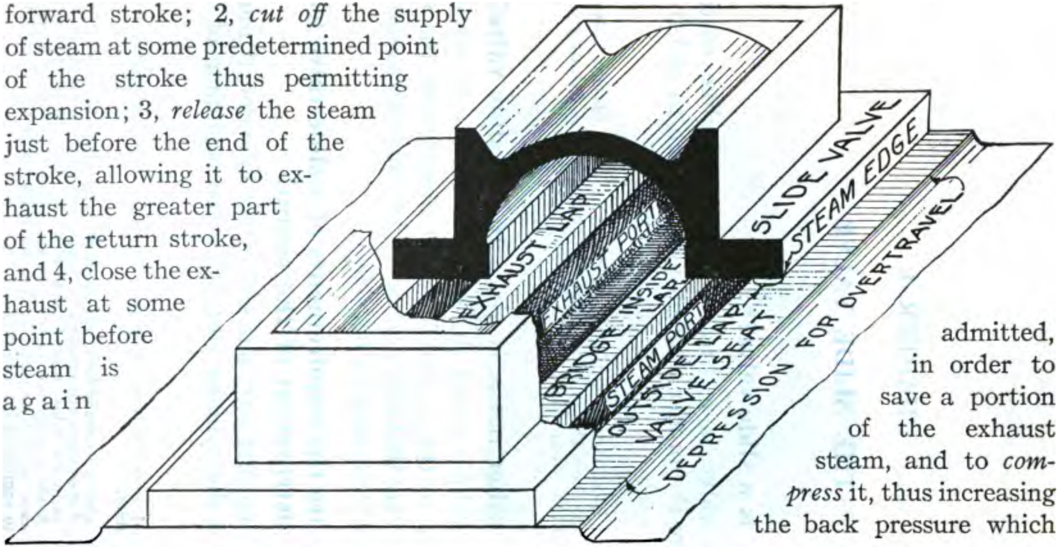


FIG. 343.—General form of the slide valve and seat. Sectional view with names of the important parts. The depression allows the valve to over travel the seat so that wear will take place uniformly.

Now considering both ends of the cylinder, it is evident that the series of events just related must be alternately performed at each end of the cylinder. For instance, while steam is being admitted

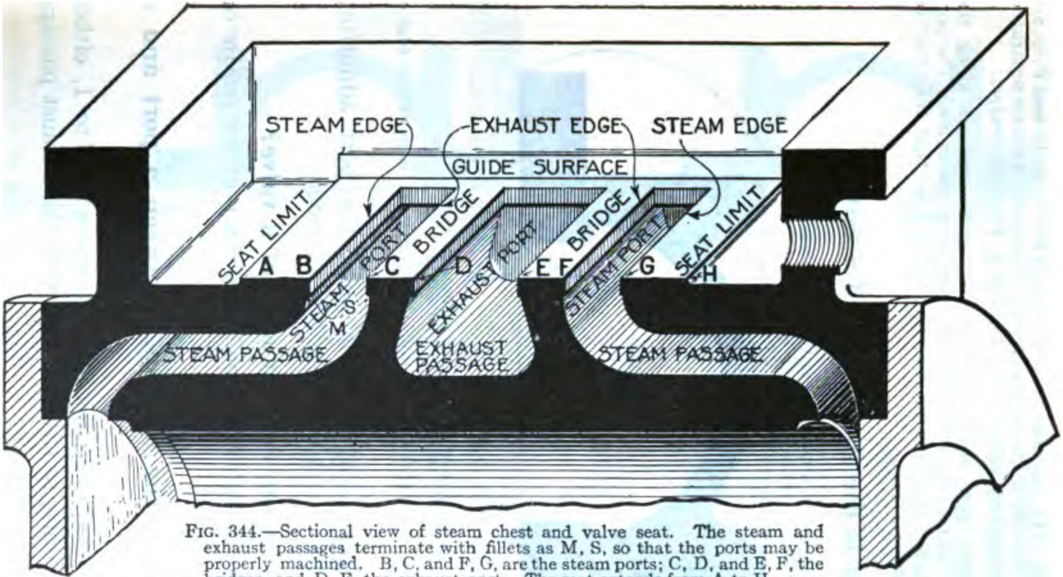


FIG. 344.—Sectional view of steam chest and valve seat. The steam and exhaust passages terminate with fillets as M, S, so that the ports may be properly machined. B, C, and F, G, are the steam ports; C, D, and E, F, the bridges, and D, E, the exhaust port. The seat extends from A to H.

and expanded in one end of the cylinder, the supply previously admitted to the other end must be exhausted, and compressed, etc.

Ques. What is a valve seat?

Ans. A flat smooth surface upon which the valve moves to and fro, *and which* is pierced by the steam and exhaust ports.

As shown in fig. 344, the seat extends from A to H. B, C, and F, G, are the steam ports, and D, E, the exhaust port. The steam ports are separated from the exhaust port by two walls C, D, and E, F, called *bridges*.

In some engines there are finished vertical guide surfaces along the two sides of the seat to prevent any side motion of the valve. The port edges are terminated by small fillets as at M and S, so that they may be properly machined.

Ques. What is the steam edge of a valve?

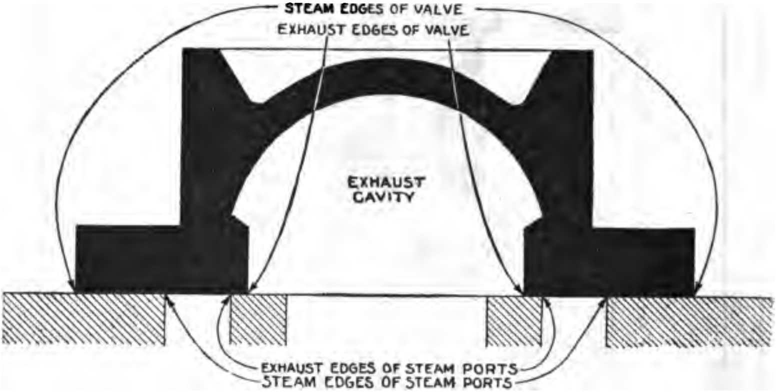


Fig. 345.—Sectional view of valve on seat illustrating the term steam edge, exhaust edge of both valve and seat.

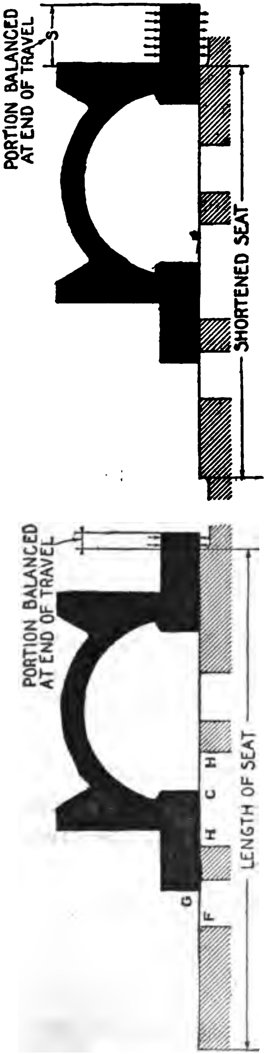
Ans. The edge which opens or closes the port admitting and cutting off the steam supply to the cylinder.

Ques. What is the exhaust edge of a valve?

Ans. The edge which opens or closes the port for *release* or *compression*.

Ques. What is the difference between a port and a passage?

Ans. A port is the entrance at the valve seat to 1, either steam passage leading to the cylinder, or 2, the exhaust passage leading to the exhaust pipe.


PORTION BALANCED AT END OF TRAVEL
S
SHORTENED SEAT
PORTION BALANCED AT END OF TRAVEL
G
F
H
C
H
LENGTH OF SEAT

FIGS. 346 and 347.—Section of seat and valve at end of travel showing effect of reducing length of seat. In fig. 346 the seat is of such length that the over travel of the valve is very small. When the valve is in the extreme position only the small portion M, will be balanced by steam acting on both sides as indicated by the arrows. If the seat length be reduced to a minimum so that when the valve is in the extreme position only enough contact area is left to produce a tight joint. The considerable portion S, which has over traveled will be balanced by the steam acting on both sides as indicated by the arrows, thus materially reducing the load on the valve, causing it to consume less power in operation and to wear less rapidly.

Ques. What governs the length (A H, fig. 344) of the valve seat?

Ans. It should be *less* than the length of the valve plus its travel so that the valve will over travel to avoid wearing shoulders in the seat.

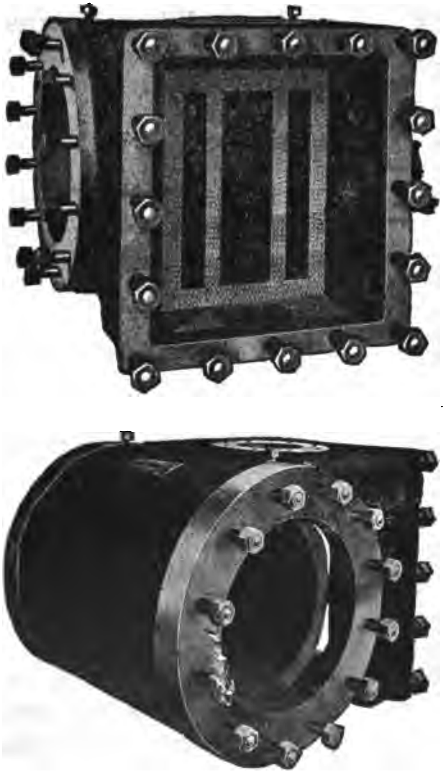
In the case of unbalanced valves the valve seat should be as short as possible to reduce the unbalancing by over travel as shown in fig. 347.

Ques. Why is the length of the ports made much greater than the width?

Ans. To secure the required amount of port opening with very little valve movement, and that the ports may be quickly opened and closed to reduce wire drawing.

Ques. What is wire drawing?

Ans. The effect produced by steam flowing through a constricted passage, which results in a fall of pressure with its attendant loss in engine operation.



FIGS. 348 and 349.—Two views of Brownell high speed engine cylinder, showing valve seat and entrance of one of the steam passages at the end of the cylinder; also arrangement of the studs, and exhaust outlet.

Ques. What governs the length of the ports?

Ans. The diameter of the cylinder; they are usually made about .8 of this diameter.

Ques. What governs the width of the ports?

Ans. For a given port length, the width depends on the amount of area required for the proper flow of the steam.

Ques. Why is the exhaust port made wider than the steam ports?

Ans. Because the valve, on account of the extent of its movement, partly covers this port during exhaust.

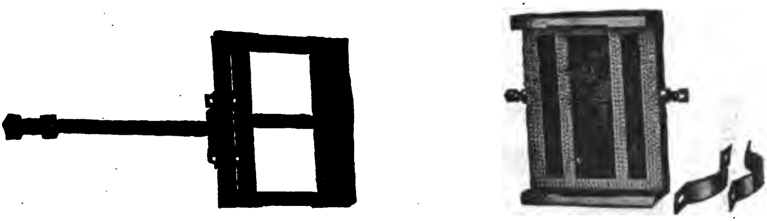
In fig. 346, the valve is shown at the extreme point of its travel, in which position the exhaust port H H′ is covered by the valve a distance H C, leaving only the opening C H′, through which exhaust steam may escape.

In a well designed valve this opening should not be less than the width F G, of the steam port. When the exhaust opening H'C, is less than the width of the steam port, the exhaust is said to be *choked* by the valve.

Ques. How is the size of the steam port obtained?

Ans. From the area and speed of the piston, and an assigned velocity of 6,000 feet per minute.*

Ques. What names are given to the principal parts of a slide valve?



FIGS. 350 to 353.—Brownell balanced slide valve and pressure plate for the cylinder shown in figs. 348 and 349. The valve is a single casting working between its seat and the pressure plate. It is double ported and is provided with means for relief from water.

Ans. The edge at either end of the valve is called the *steam edge* as shown in fig. 345, because this edge controls the admission of steam to the cylinder, and for a similar reason each inner edge is called the exhaust edge.†

---

*NOTE.—In engines having separate exhaust ports, the steam ports are proportioned for a velocity of 8,000 feet per minute, but when the same port is used for both admission and exhaust, the port must evidently be proportioned with respect to the exhaust, that is the steam should be exhausted at less velocity than the velocity of admission. In the exhaust pipe the velocity should be still less than in the steam passage—usually 4,000 feet per minute.

†NOTE.—These terms should be remembered as well as the similar names given to the edges of the steam ports as shown in fig. 345.

The area of the steam port is found as follows:

$$\text{Area steam port} = \frac{\text{area piston in sq. ins.} \times \text{piston speed in ft.}}{6{,}000}$$

Ques. What important defect is there in the operation of the ordinary slide valve?

Ans. The excessive pressure caused by the steam pressing the valve against its seat causing considerable friction and wear.*

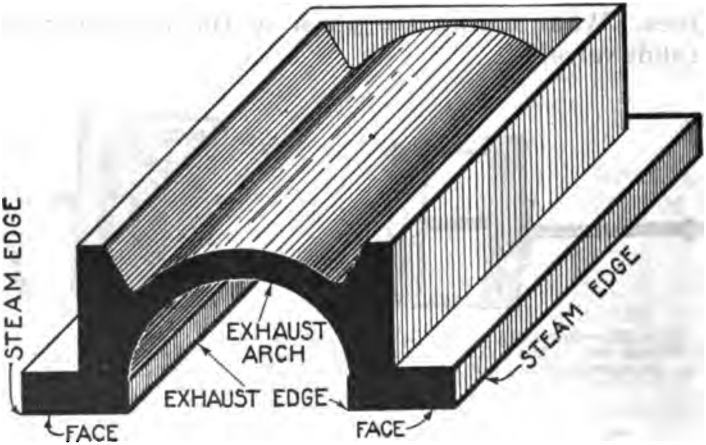


Fig. 354.—Sectional view of slide valve showing the principal parts. It is important to remember the names given in the figure.

Example.—What will be the force required to move a 9×18 slide valve, if there be on it an unbalanced steam pressure of 140 pounds per square inch, and the resistance due to friction be .02 of the total load on the valve?

---

*NOTE.—A slide valve whose outside dimensions are, say, 9×18 inches has an area of 162 square inches. If a boiler pressure of 140 pounds per square inch be exerted on this area, then the total pressure on the area is 162×140=22,680 pounds. The actual pressure which tends to force the valve against its seat is variable, as during some portions of the stroke the steam in the ports under the valve exerts an upward pressure, which opposes that on top. The pressure on top is also influenced by the fit of the valve. If it be not steam tight, more or less steam will get between the valve and its seat, and thus act against the pressure on top, whereas if the valve be steam tight, no such action will occur. In any event the pressure on top of an unbalanced valve is very considerable.

Area of valve = 9 × 18 = 162 sq. ins.

Total load on valve due to steam pressure:

= 162 × 140 = 22,680

Force required to move the valve:

= 22,680 × .02 = 453.6 pounds

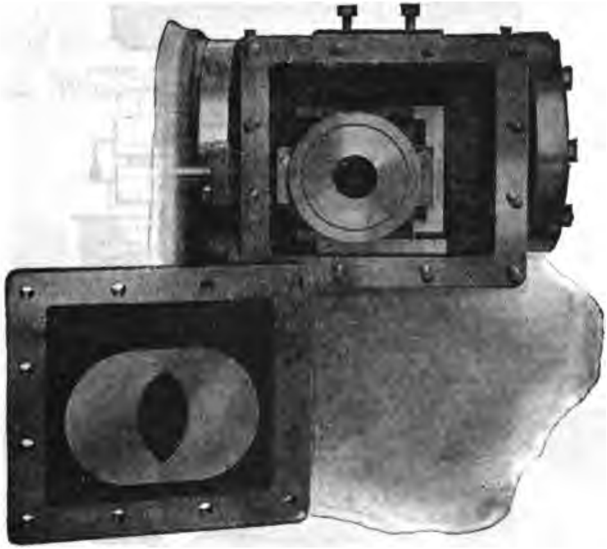


FIG. 355.—Cylinder of Houston, Stanwood and Gamble automatic engine with valve chest cover removed showing balanced slide valve. Fig. 356 shows a sectional view of this valve.

Ques. What provision is sometimes made for relieving the pressure on the slide valve?

Ans. Various devices have been used to exclude steam from the top of the valve, so that the pressure cannot be exerted in a direction which would press the valve against its seat.* The valve is then said to be *balanced*.

---

*NOTE.—Experiments with small engines show that from one to two per cent of the whole power of the engine is absorbed in moving the slide valve when unbalanced.

Lap

Ques. What is the lap of a valve?

Ans. It is that portion of the valve face which overlaps the steam ports when the valve is in its central position.

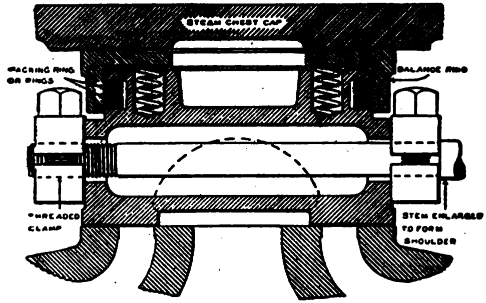


Fig. 356.—Sectional view of Houston Stanwood and Gamble balanced slide valve. It is held against the seat by steam pressure on a small area of the back of the valve and by this means is made nearly steam tight. Also, the valve by this means is made to automatically follow up its wear, so that the steam consumption will be lower after the engine has been in use for a time, rather than higher.

In fig. 357 AB, is the *outside* or *steam lap*, and CD, the *inside* or *exhaust* lap.†

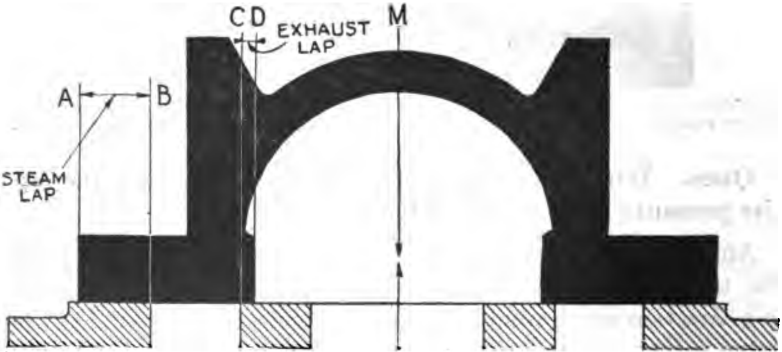


Fig. 357.—The plain D slide valve showing "lap." A B is the *outside* or *steam lap*; **C D**, the *inside* or *exhaust lap*. The figure also illustrates the *neutral position* of the valve.

---

†NOTE.—Since, in some types or engine, steam is admitted at the inside edges of the ports and exhausted at the outside edges, the terms *steam lap* and *exhaust lap* are therefore sometimes used to avoid confusion.

Ques. What is usually understood by the term lap?

Ans. The outside or steam lap.

Ques. What is the effect of lap?

Ans. It causes the valve to shut off the supply of live steam to the cylinder before the end of the stroke, the greater the amount of lap the sooner does this occur.

Ques. What is the effect of inside lap?

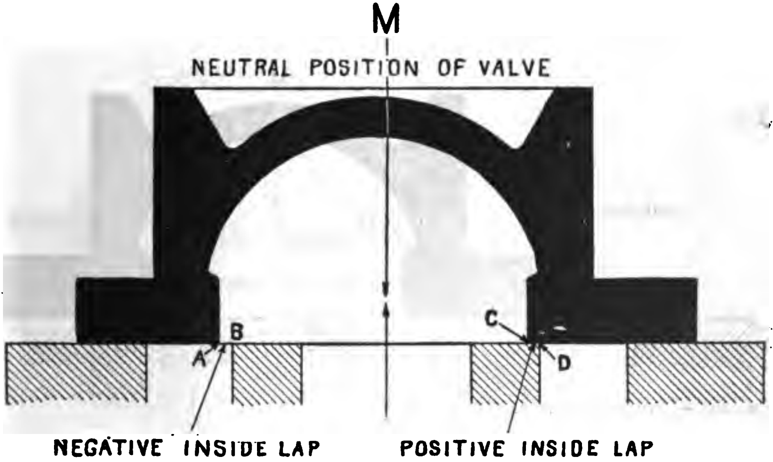


FIG. 358.—Illustrating *negative* and *positive inside lap*. A valve is sometimes given negative inside lap at one end to equalize release and compression; the irregularity being due to the angularity of the connecting rod.

Ans. It causes the valve to open later for exhaust and close sooner, thus shutting in a larger portion of the exhaust steam.

Ques. What is the neutral position of a valve?

Ans. Its central position, or the mid-point of its travel as shown in fig. 358, overlapping, equally the steam ports. In this position a line drawn through the center of the valve will coincide with a line drawn through the center of the exhaust port, when the exhaust lap is the same at each end.*

Ques. What is negative inside lap?

Ans. The space (as A B, fig. 358) sometimes left at one end of the valve between its exhaust edge, and the exhaust edge of the steam port when the valve is in its neutral position.

It is sometimes, though ill advisedly, called *inside clearance.* The difference between *negative* and *positive*† inside lap is indicated in the figure.

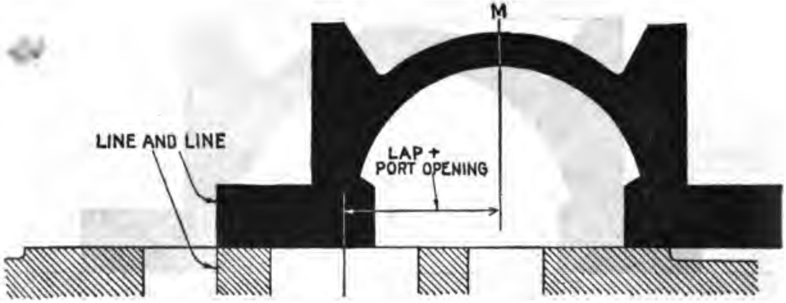


Fig. 359.—Illustrating "line and line" position.

Ques. Why is a valve sometimes given negative inside lap at one end?

Ans. To equalize certain irregularities of exhaust due to the *angularity*‡ of the connecting rod.

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*NOTE.—These center lines are convenient in locating the valve for different positions.

†NOTE.—The term inside lap unqualified always means *positive* inside lap.

‡NOTE.—The term angularity and its effect on the action of the valve gear is later fully explained.

Ques. What is the effect of negative inside lap?

Ans. It causes the valve to open sooner and close later to exhaust.

That is, *pre-release* begins earlier, and *compression* later.

Ques. What is "line and line" position?

Ans. When one edge of the valve is in the same line or plane with the corresponding edge of the port as in fig. 359.

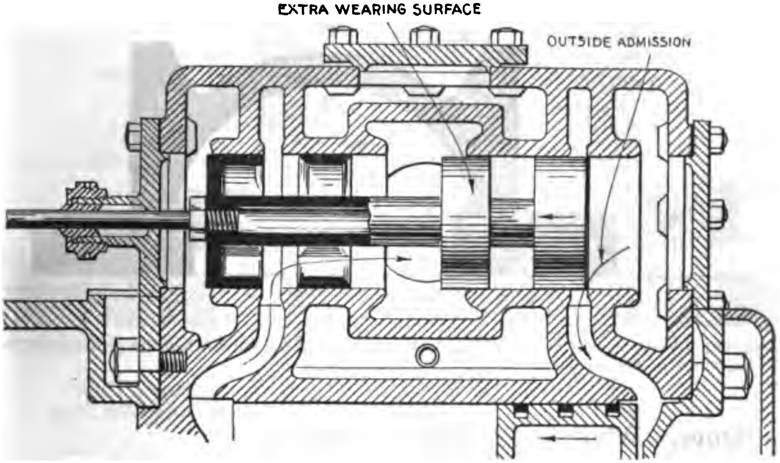


Fig. 360.—Sectional view of Phoenix cylinder showing double disc piston valve. The valve admits steam from the ends, the valve casing being surrounded by live steam. The two inner discs provide extra wearing surface.

Here the steam edge of the valve is in line with the steam edge of the port, and valve is at the point of opening the port.

Lead

Ques. What is lead?

Ans. The amount by which the port is open for the admission of steam when the piston is at the beginning of the stroke.*

For instance, in fig. 361, the port is open a distance A A′, which is the lead. This is *outside lead* as distinguished from *inside lead*. It is also called *positive lead* to distinguish it from negative lead.

Ques. What is negative lead?

Ans. The amount by which the steam port is closed to admission when the piston is at the beginning of the stroke.

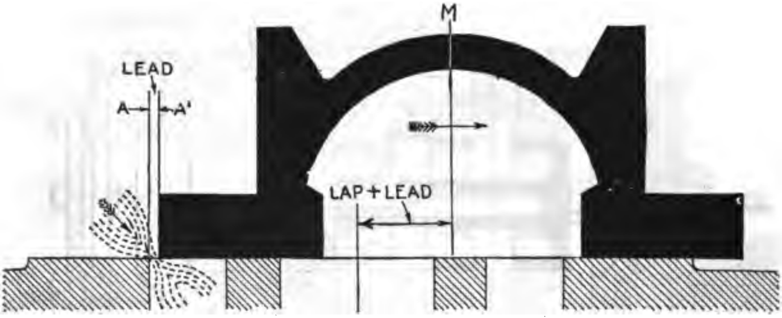


FIG. 361.—Valve in *lead position*. This is the position of the valve when the piston is at the beginning of the stroke.

Ques. What is the object of lead?

Ans. Lead is given to a valve in order to admit live steam to the cylinder before the beginning of the stroke so that the pressure of the compressed exhaust steam in the clearance

---

*NOTE.—Lead varies with the size and type of engine usually from zero to about ⅜ of an inch. Small or medium sized slow speed engines may have from $^1/_{64}$ to $^1/_{16}$, medium speed engines a greater amount, and in the case of high speed engines still more unless there be considerable compression. Vertical engines usually have more lead at the lower cylinder end than at the upper, in order to assist the compression in resisting the excess momentum at the lower end due to the weight of the moving parts acting in that direction. ***In general,*** the greater the compression, the less the amount of lead required. The principal object of lead is to secure full steam pressure in the cylinder at the beginning of the stroke, rather than to resist the momentum of the moving parts, although in high speed engines where the compression is not sufficient to bring the moving parts to a state of rest, an extra amount of lead necessary on this account.

space will be increased to boiler pressure at the beginning of the stroke. This enables the piston to begin its stroke with the maximum pressure.

Ques. What is the object of negative lead?

Ans. On some types of valve gear, as the link motion, the lead increases with the degree of expansion. Hence, in full gear, that is, for the maximum cut off, a negative lead is given to prevent excessive positive lead when cutting off very early.

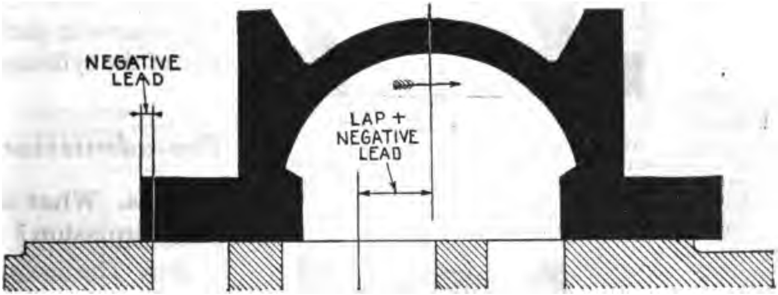


FIG. 362.—Valve in *negative lead position.* Negative lead is sometimes given with link motion full gear to prevent excessive lead when cutting off short, as on locomotives.

With negative lead, the valve does not open for the admission of steam until *after* the beginning of the stroke.

Ques. What is inside or exhaust lead?

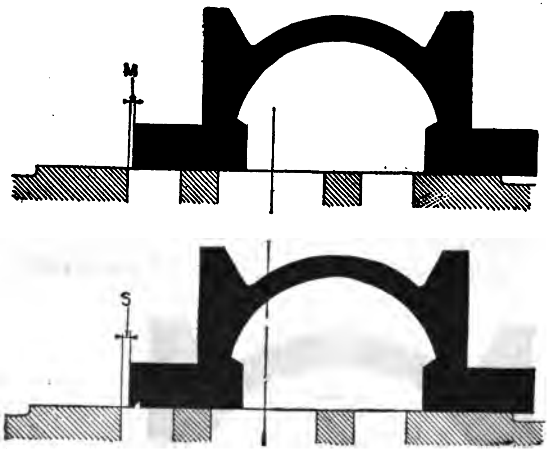
Ans. It is the amount by which the steam port is opened to exhaust when the piston is at the beginning of the stroke.

Ques. What is constant lead?

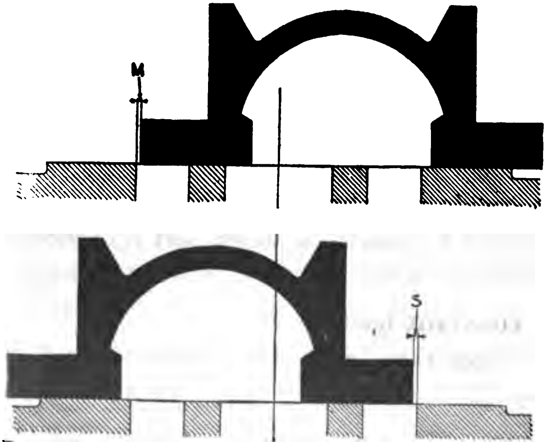
Ans. Lead which does not change for different degrees of expansion.

---

*NOTE.—Negative lead is sometimes given to the valves of express locomotives when fitted with link motion valve gear. Since the action of the link increases the lead for an early cut off (the cut off used except at starting) negative lead is given for full gear so that the lead will not be excessive at early cut off.



FIGS. 363 and 364.—Valve in lead position illustrating *variable lead;* thus lead M, with gear in late cut off position is less than lead S, for early cut off. A peculiarity of the link motion gear is this variable lead. With the link motion in shortening the cut off by "hooking up"; *open rods* give *increasing lead,* while *crossed rods* give *decreasing lead.*



FIGS. 365 and 366.—Valve in lead positions illustrating *equal lead,* that is, lead M at one end of the cylinder is the same as lead S at the other end.

Ques. What is variable lead?

Ans. Lead which changes with the degree of expansion.

Ques. What is equal lead?

Ans. Lead which is the same at each end of the cylinder.

Pre-admission

Ques. What is pre-admission?

Ans. The flow of live steam into the cylinder *before* the beginning of the stroke.

Ques. On what does pre-admission depend?

Ans. On the amount of lead.

That is, the greater the lead, the sooner does the valve open to admit steam before the beginning of the stroke.

Ques. What is the object of pre-admission?

Ans. Principally to secure the full steam pressure at the beginning of the stroke, and in addition, in some cases, to assist the *compression* in bringing the moving parts to a state of rest.

Ques. What objection is there to pre-admission?

Ans. It increases the period during which live steam, at a high temperature, is exposed to the comparatively cool cylinder walls, thus tending to increase *initial condensation*.

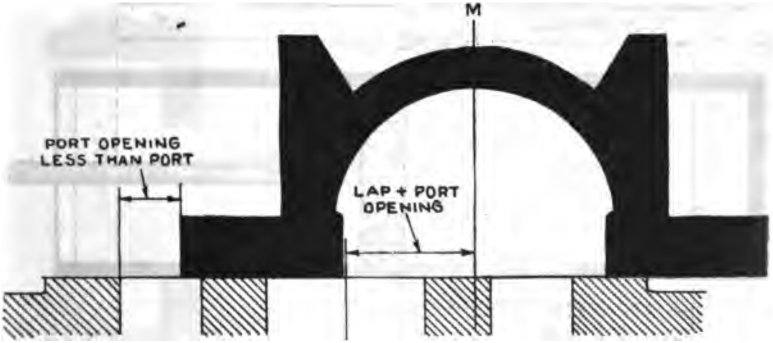


FIG. 367.—Valve fully opened for admission; this comes when the piston is about half way between the beginning of the stroke and cut off position. Usually the port is only partially opened, as here shown, because the speed of the steam through the port during admission should be greater than during exhaust.

NOTE.—***In the design*** of a steam engine it is important to give the proper amount of port opening, for if it be too small. the velocity of the entering steam is unduly increased, resulting in a loss of pressure commonly known as ***wire drawing***. On ***fixed cut off*** engines, the port opening may be less than the port because the latter is proportioned to give the proper velocity of the exhaust steam. With ***variable cut off*** gears, whose action reduces the port opening in shortening the cut off, as for instance the link motion, it is usual to provide excess port opening at full gear in order to obtain adequate admission when linked up. For instance on locomotives which operate normally at short cut off, the port opening in full gear is considerable, even exceeding the width of the port as in fig. 375.

NOTE.—Engines are commonly designed with ports and passages proportioned for a ***nominal steam speed of 6,000 to 8,000 feet per minute when cutting off at about 60% of the stroke***, using, instead of the maximum velocity. the average velocity, that is

$$\textit{area port} = \frac{\textit{area piston} \times \textit{piston speed}}{6{,}000 \text{ to } 8{,}000}$$

the port and piston areas being taken in sq. ins., and the piston speed in feet per minute.

NOTE.—***It must be obvious*** that in determining the port opening, the cut off should be taken into consideration, since the movement of the piston is not uniform but starts from zero at the beginning of the stroke and gradually accelerates to maximum velocity near mid stroke, then decreases to zero at the end of the stroke. Prof. Fessenden in his excellent book on valve gears has given an elaborate treatment of this subject which should interest those engaged in engine design.

Ques. What is meant by initial condensation?

Ans. The condensation of live steam in the cylinder which takes place during the periods of pre-admission and admission.

Admission

Ques. What is admission?

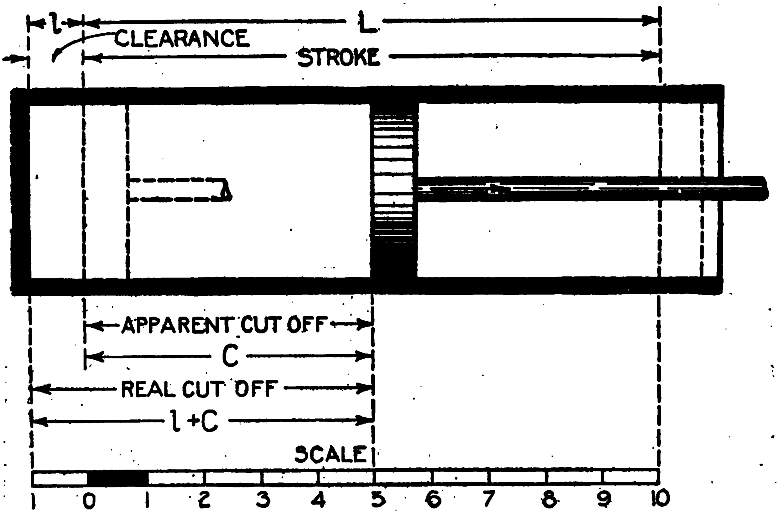


FIG. 368.—The *apparent* and *real* cut off. The effect of cylinder clearance is to make the number of expansions less than would correspond to the apparent cut off, that is, the cut off of the valve gear. Thus, if the valve gear cut off at one-half stroke, there would be without clearance, two expansions. With, say 10 per cent clearance, the expansions would be reduced to 1÷(.1+.5)=1.66. The real cut off would then be 1÷1.66=.6 stroke. In the figure the clearance volume *l* includes besides the volume between the piston at end of stroke and cylinder head, the volume of the steam passage (not shown) up to the steam port.

Ans. The flow of *live steam** into the cylinder from the beginning of the stroke to the point of cut off.

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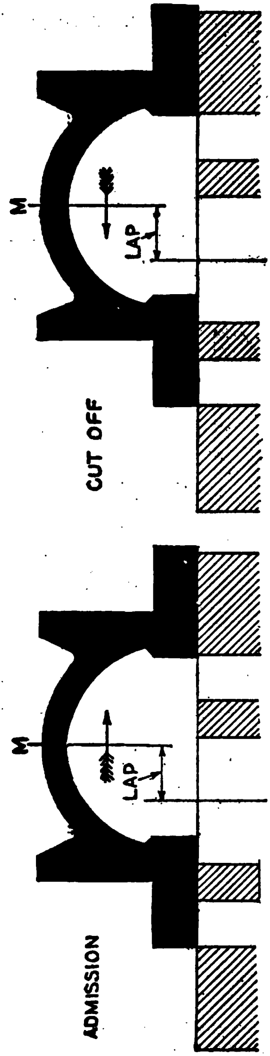
*NOTE.—Live steam is steam taken direct from the boiler, and which has not been expanded in the cylinder, as distinguished from steam which has been admitted to the cylinder and expanded in doing work on the piston.



FIGS. 369 and 370.—Positions of the valve at the beginning of admission and at cut off. It will be noted that the position of the valve is the same in each case but the direction of motion is not the same.

Cut off

Ques. What is cut off?

Ans. It is the closure of the steam port to the admission of steam.

Ques. How is it usually expressed?

Ans. As a fraction of the stroke, as one-half, five-eighths, or three-fourths cut off.

Ques. What is cut off thus expressed called?

Ans. The apparent cut off.

Ques. Why?

Ans. Because it does not represent the actual point at which cut off takes place when clearance is considered.

As distinguished from the apparent cut off, the term *actual* or *real cut off* is used to indicate the point at which the steam port is closed, taking clearance into account.

Ques. What is the real cut off?

Ans. The sum of the apparent cut off plus the percentage of clearance.

For instance, if the apparent cut off be one-half, or 50 per cent and the clearance be 10 per cent, then the real cut off is 10+50=60 per cent of the stroke.

In fig. 368, let the volume *l*, at the end of the cylinder represent the clearance in proportion to the volume displaced by the piston during the stroke L. This clearance* volume includes all the space between the piston when it is at the beginning of the stroke and the face of the valve* plus the volume of the steam passage up to the steam port.

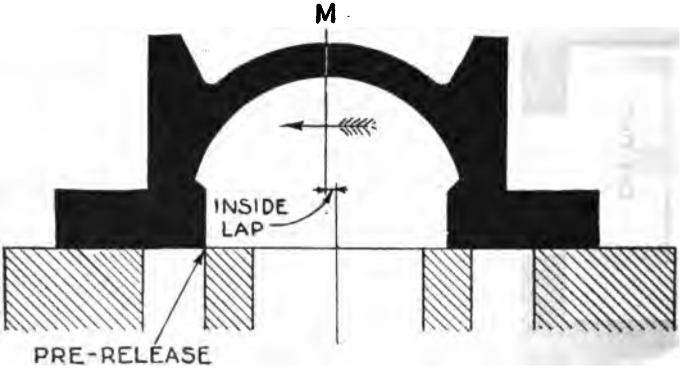


FIG. 371.—Position of valve at beginning of *pre-release*. This occurs just before the piston reaches the end of the stroke so as to rid the cylinder of most of the steam before the beginning of the return stroke, and thus reduce the back pressure of exhaust as much as possible.

It is evident from the figure that the distance the piston has moved from the beginning of the stroke does not represent the real cut off, and that the latter is made up of the volume displaced by the piston plus the clearance volume, or *l*+C. Thus in the figure if cut off occur when the piston is at 5, as shown, the apparent cut off C, is five-tenths or one-half stroke. The actual cut off *l*+C, is six-tenths stroke.

Pre-release

Ques. What is pre-release?

---

*NOTE.—The term clearance is sometimes used to denote the linear distance between the piston and the cylinder head when the piston is at the beginning of the stroke.

Ans. The opening of the steam port to exhaust before the piston has completed its stroke, as shown in fig. 371.

If the steam were confined in the cylinder until the piston had reached the end of its stroke, there would not be time for it to escape without creating considerable back pressure.

Ques. On what does pre-release depend?

Ans. Primarily upon the amount of exhaust lap, and in design, upon the conditions of operation.

The proper amount of pre-release depends on the piston speed, and the quantity of steam to be discharged. Usually the valve is so proportioned that pre-release begins about 90 per cent. of the stroke.

Release

Ques. What is release?

Ans. The exhaust of steam from the beginning of the return stroke to the point of compression.

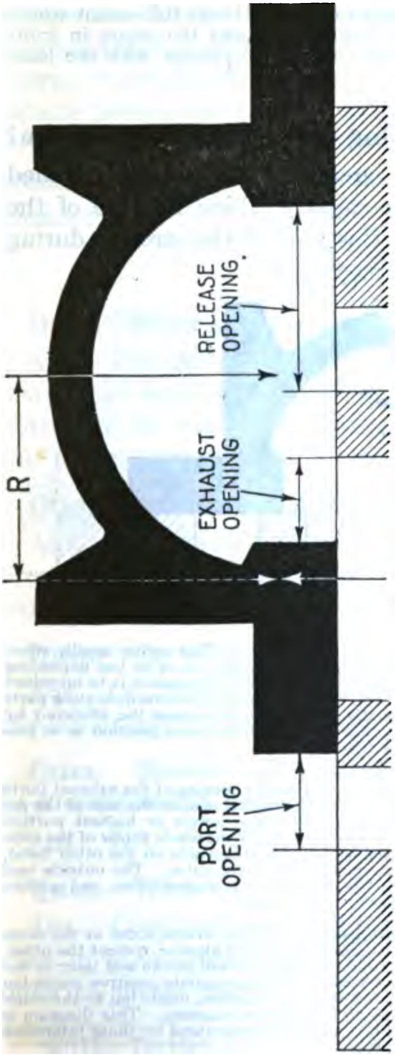


FIG. 372.—Valve in full release position, that is, in the position of maximum opening to exhaust, this occurs when the piston is nearly half way between the beginning of the return stroke and the compression position.

In fig. 372, the valve is shown open to exhaust to its full extent which occurs during this period, and at this instant allows the steam in front of the fast advancing piston to escape from the cylinder with the least possible back pressure.

Ques. What happens during pre-release and release?

Ans. During pre-release the greater part of the expanded steam is exhausted, the pressure rapidly falling because of the slow movement of the piston at this part of the stroke; during

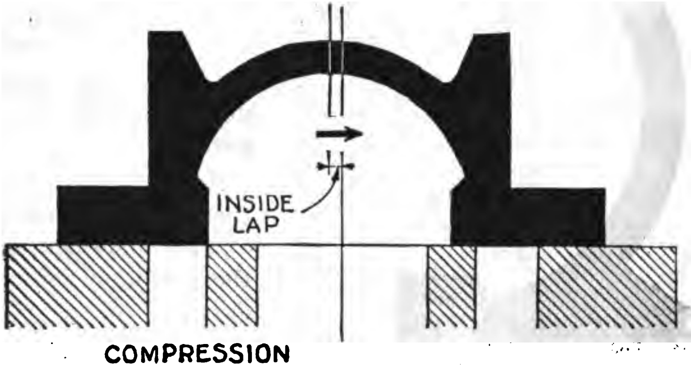


FIG. 373.—Position of the valve at the beginning of compression. This occurs usually when the piston has travelled about three-quarters of the return stroke, more or less depending upon the type of engine and working conditions. The object of compression is to introduce a spring like back pressure to absorb or "cushion" the momentum of the reciprocating parts and bring them to a state of rest at the end of the stroke; also to increase the efficiency by saving some of the exhaust steam. Note that the valve is in the same position as at pre-release (fig. 371) but is moving in the opposite direction.

NOTE.—***Inside Lead:*** "Experiments show that the earlier opening of the exhaust ports is especially of advantage, and in the best engines the lead of the valve upon the side of the exhaust or the inside lead is $^1/_{25}$ to $^1/_{15}$, *i. e.*, the slide valve at the lowest or highest portion of the piston has made an opening whose height is $^1/_{25}$ to $^1/_{15}$ of the whole throw of the slide valve. The outside lead of the slide valve or the lead on the steam side on the other hand, is much smaller, and is often only $^1/_{100}$ of the whole throw of the valve. The outside lead of the slide valve or the lead on the steam side on the other hand, is much smaller, and is often only $^1/_{100}$ of the whole throw of the valve—*Weisbach* (vol. ii, p. 296).

NOTE.—***Equalized Pre-Release and Compression.*** These events occur at the same time when the valve has no inside lap and the correction of one will likewise correct the other. It is desired to cause both these events to occur earlier in the forward stroke and later in the return stroke. To accomplish this, it is necessary to give an appropriate positive inside lap to the end of the valve nearest the crank shaft, and an equal negative, inside lap to the other end. The laps are easily determined by means of the Bilgram diagram. This diagram is explained at length in this Chapter, and should be thoroughly understood by those interested in valve gears.

release the exhaust pressure is always a little higher than the external pressure (that is, higher than the pressure of the atmosphere, or condenser as the case may be) on account of the rapidly advancing piston forcing the steam through the restricted passage, port and exhaust pipe at great velocity. This pressure is called the *back pressure of exhaust*, or simply back pressure.

Compression

Ques. What is compression?

Ans. The closure of the steam port to exhaust before the piston has reached the end of its stroke, thus shutting in a portion of the exhaust steam, and by the forward movement of the piston, compressing it until pre-admission begins.

Ques. What is the effect of compression?

Ans. The rapidly increasing back pressure due to compression acts as a spring to *cushion* the momentum of the moving parts, and brings them to a state of rest at the end of the stroke.*

Ques. On what does compression depend?

Ans. On the exhaust lap and angular advance; the greater the lap or angular advance the sooner compression begins.

Ques. Should compression begin at the same point in a condensing engine as in a non-condensing engine?

Ans. No.

Ques. Why?

Ans. Compression should begin earlier in a condensing engine than in a so called high pressure or non-condensing engine in order to get an equal amount of cushioning.

---

*NOTE.—The effect of compression is sometimes called *cushioning*.

The compression curve of a condensing engine, because exhaust takes place at a lower pressure, does not rise so rapidly as when running non-condensing, hence, for equal cushioning, compression must begin sooner than in a non-condensing engine.

Ques. Does compression result in a loss of energy?

Ans. No, because the power required to compress the confined steam is again given out by its expansion behind the piston on the next stroke. In fact, there is a direct saving, as the compressed steam is utilized to help fill the clearance space instead of filling it entirely with live steam.

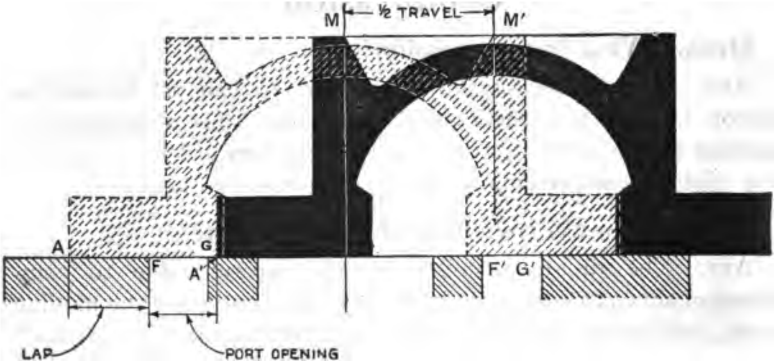


Fig. 374.—Illustrating *port opening* and *half travel* of the valve. The valve is shown in dotted section in its neutral position, and in full section in its *extreme position*. Half travel *is equal to the lap plus the port opening;* the latter being the distance the steam edge of the valve moves past the steam edge of the port during admission. This represents the movement of the valve on either side of its central or neutral position and =*lap*+*port opening*=A F +F A′. As drawn, the port opening is greater than the port by the amount G A′.

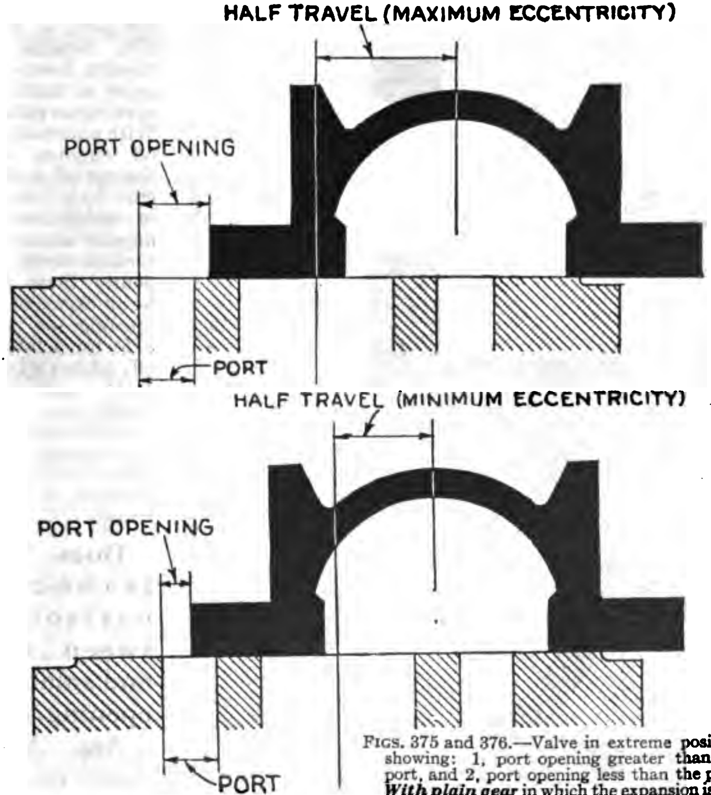
Port Opening

Ques. What is port opening?

Ans. The extent to which the steam port is opened when the valve is at the end of its to and fro movement as F A′, fig. 374.

Ques. What is the relation between the port opening and the width of the port?

Ans. It may be either greater or less than the width of the port, as shown in figs. 375 and 376.


HALF TRAVEL (MAXIMUM ECCENTRICITY)
PORT OPENING
PORT
HALF TRAVEL (MINIMUM ECCENTRICITY)
PORT OPENING
PORT

FIGS. 375 and 376.—Valve in extreme position showing: 1, port opening greater than the port, and 2, port opening less than the port. ***With plain gear*** in which the expansion is not variable, the valve is designed for a port opening less than the port, because for admission less port area is required than for exhaust, the steam being admitted in good practice, at a velocity of 8,000 ft. per minute and exhausted at 6,000 ft. per minute. ***With variable expansion gears,*** which vary the expansion, as later explained by the method of ***combined variable throw and variable angular advance***, the travel of the valve is considerably reduced for early cut off, hence, the port opening is made more than sufficient at late cut off as in fig. 375, in order that the reduced opening for early cut off as in fig. 376 will not be too small.

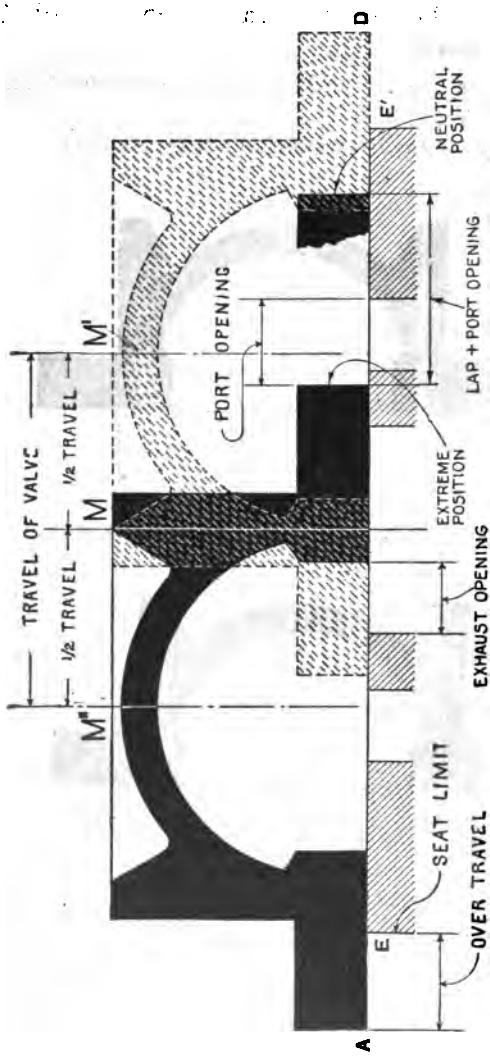


FIG. 377.—Showing valve in the two extreme positions M' and M'', to illustrate the *travel*. M, half way between corresponds to the neutral position. The valve as shown moves to either side of M, a distance equal to the *lap plus the port opening*; the *travel* or entire movement, therefore, ***is equal to twice the lap plus twice the port opening.***

In the case of a throttling engine where the travel of the valve does not change, the port opening may be less than the width of the port, as in fig. 376, because admission does not require as much port opening as exhaust. With automatic cut off engines, when the cut off is shortened by the method of ***combined variable angular advance and variable throw***, evidently there must be an excess of port opening (as in fig. 375) for a late cut off, otherwise at early cut off it would be insufficient for admission, on account of the considerable decrease in valve travel.

Ques. What is the comparison between large and small port opening?

Ans. A large port opening permits very early cut off without choking the admission.

On account of the greater velocity of the valve, the events of the stroke such as cut off release, etc., are more sharply defined; in other words, there is less wire drawing. These features are offset somewhat by the increased wear of the valve and larger valve gear necessary to secure the increased valve travel.

Travel

Ques. What is the travel* of a valve?

Ans. The extent of its to and fro movement as shown in fig. 377.

Here the valve is shown in full lines at one end of its travel, and in dotted lines at the other end, the travel being the distance M″ M′.

Ques. How is the travel obtained?

Ans. From the lap and the port opening.

Travel of valve = twice the lap + twice the port opening.

In fig. 374, the valve is shown in dotted lines in its neutral position M and in full lines at one end of its travel M′.

It is evident from the figure that the valve has moved a distance to the right equal to the *lap* A F, plus the *port opening* F A′.

Now to admit steam to the other end of the cylinder through the port F′ G′, the valve must move an equal distance to the left of its neutral position M, that is a distance equal to M M″ in fig. 377. Hence, the travel *equals twice the lap plus twice the port opening.* The full travel M″ M′ is shown in fig. 377.

Ques. What is over travel†?

---

*NOTE.—In the year 1836, the word *travel* was used in a different sense from the present meaning. According to Wansbrough, in order to keep the steam on the piston as long as possible, the valve moved nearly one-half inch beyond the port at each end or "over opened"—this distance or movement beyond the port was called the travel.

†NOTE.—The term *over travel* is used by some writers to denote the distance the steam edge of the valve moves beyond the exhaust edge of the steam port in opening it. The author prefers to define it with respect to the seat limit.

Ans. The extent to which the steam edge of the valve moves beyond the seat limit, as A E or E′ D, fig. 377.

Ques. What is the object of over travel?

Ans. To preserve uniform wear of the seat, and reduce the unbalanced load on the valve.

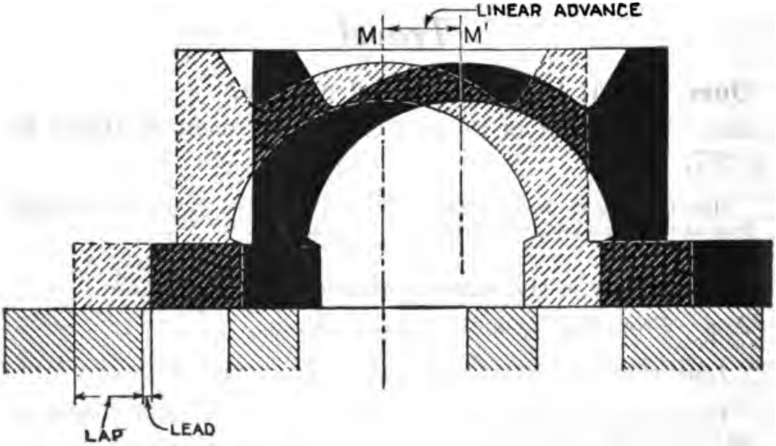


Fig. 378.—Showing valve in its position of *linear advance.* When the piston is at the beginning of the stroke, the valve must be at a distance from its neutral position equal to the *lap plus the lead.* The valve is shown in solid black in its linear advance position, and in dotted section in its neutral position.

Linear Advance

Ques. What is the linear advance of a valve?

Ans. It is the distance the valve has moved from its neutral position, when the piston is at the beginning of the stroke.

Ques. Upon what does linear advance depend? *

---

*NOTE.—It should be remembered that the word lap unqualified always means outside or steam lap. The same is true of lead.

Ans. On the amount of lap and lead.

Thus in fig. 378, the valve has moved from its neutral position a distance M M′, equal to its linear advance. The valve in its neutral position is shown in dotted lines and in its linear advance position in full section.

From the figure it is clear that:

linear advance = lap* + lead.

Early Cut Off

Ques. How may the cut off be varied?

Ans. By changing **both** *the angular advance and throw of the eccentric;* the greater the angular advance and shorter the throw, the earlier the cut off.

In order not to unduly affect the other events of the stroke, the valve gear, as will be later explained, is arranged to increase the angular advance simultaneously as the throw is reduced, this being called the method of ***combined variable angular advance and variable throw.***

Ques. What objection is there to shortening the cut off by the above method?

Table Showing Effect of Changes in Lap, Throw, Angular Advance, Etc.

Event	Period	Increasing *outside lap*	Increasing *inside lap*	Increasing *angular advance*	*Increasing throw*	Increasing *angular advance*, decreasing *throw*			
						Shifting eccentric	*Swinging eccentric*		
							Pin on crank	Pin opposite crank	Offset pin on crank
lead..........	....	reduced	unchanged	increased	increased	constant	increased	decreased	small increase
....	pre-admission	reduced	unchanged	increased	increased	unchanged	increased	decreased	small increase
port opening...	....	reduced	unchanged	unchanged	increased	reduced	reduced	decreased	decreased
cut off.........	....	earlier	unchanged	earlier	later	earlier	earlier	earlier	earlier
....	admission	reduced	unchanged	unchanged	increased	reduced	reduced	reduced	reduced
....	expansion	increased	increased	unchanged	reduced	increased	increased	increased	increased
pre-release	....	unchanged	later	earlier	unchanged	earlier	earlier	earlier	earlier
....	exhaust	unchanged	reduced	unchanged	unchanged	unchanged	unchanged	unchanged	unchanged
exhaust opening	....	unchanged	reduced	unchanged	increased	reduced	reduced	reduced	reduced
compression	....	unchanged	earlier	earlier	unchanged	earlier	earlier	earlier	earlier

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*NOTE.—See note on page 206.

Ans. The shorter the travel, the less the port opening, hence for very early cut off there is insufficient port opening for admission, moreover, pre-release begins earlier and compression later.

Ques. How may the first objection be overcome?

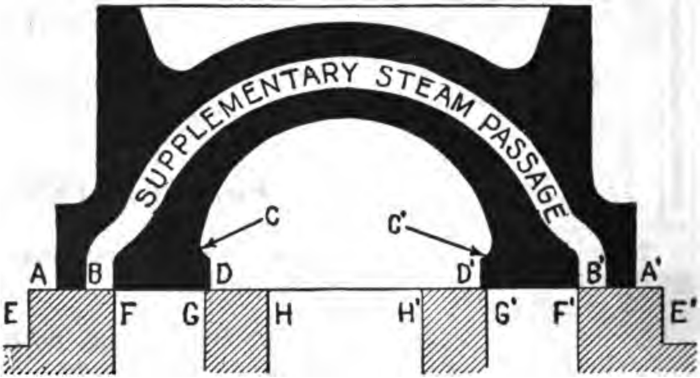


Fig. 379.—The Allen valve. The supplementary passage is for double admission which is desirable on locomotives fitted with link motion as they are usually run with short cut offs, and the action of the link under these conditions gives very little port opening.

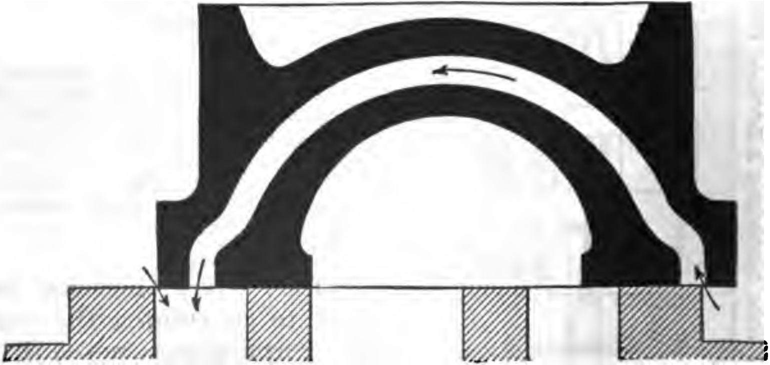
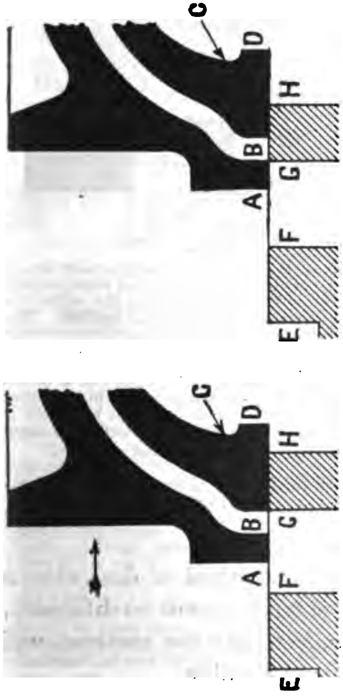


Fig. 380.—The Allen valve in lead position showing double admission. It should be noted that the second admission through the supplementary port, depends on the length of the seat which forms the lap.

Ans. By providing a *supplementary passage* in the valve for the admission of steam, as shown in fig. 379, which represents the Allen valve, used principally on locomotives. This passage terminates in the lap at B F and B′ F′. The seat limit is such that A F, the lap of the valve equals B′ E′, which acts as lap for the supplementary passage. As the valve moves, for instance, to the right, A, passes F, at the same time that B′, passes E′, hence, steam is admitted simultaneously by both ends of the valve as shown in fig. 380. The effect is the same as an ordinary valve moving at double speed. Following the further movement of the valve, it will be seen that the valve continues to open the port FG at,



FIGS. 381 and 382.—The maximum admission of the Allen valve is equal to the width of the port F G (fig. 381), less the width of the valve bridge A B, because as the valve moves to its extreme position (fig. 382) the supplementary port B G, is closed by the valve seat.

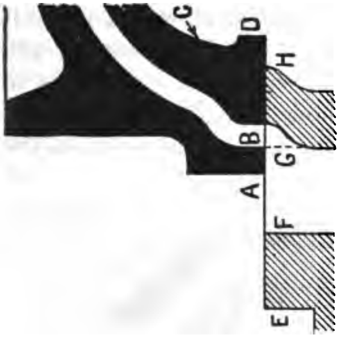
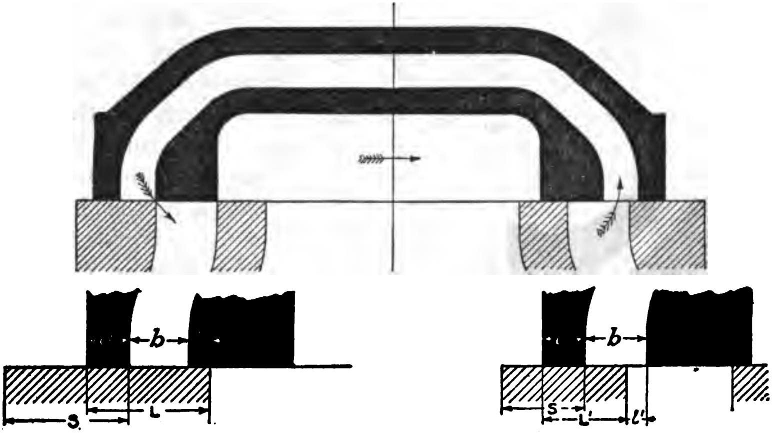


FIG. 383.—The Allen valve with modified steam ports. The steam ports are so modified that the supplementary port is not closed when the valve is in the extreme position. The amount of enlargement of the steam port for proper admission will depend on the width of the valve bridge A B.

double speed until it has reached the position shown in fig. 381. Here the total port opening is F A+B C. This opening will not be increased by any further movement of the valve in traveling to its extreme position as shown in fig. 382, on account of the closing of the supplementary port. In this position the opening F A (fig. 382) = F A (fig. 381) + B G.



Figs. 384 to 386.—Allen valve with negative lap. When the valve has a large steam lap, it is not difficult to keep the opening of supplementary port within this lap, and even to leave a small positive inside lap *l*, fig. 385, but when the main lap *s* is smaller, or the width *b* is increased, there will be a negative lap to the supplementary port as at *l*, fig. 386. The result is, that during a brief time a passage is opened from one end of the cylinder to the other, with some tendency to modify the exhaust operation. Thus in fig. 386 the valve is at the distance *l'* and moving toward the right. Compression has begun in the left end of the cylinder, and now steam which has not yet been released in the right end is about to flow over and increase the pressure and the amount of the clearance steam in the other end. This action will occur while the valve travels through the distance 2*l*, but since it moves rapidly near neutral position and the openings involved are small, the effect upon the steam distribution will be small.

If the steam ports be enlarged as shown in fig. 383 they would be completely opened when the valve reaches the extreme position. This would, however, necessitate a somewhat longer valve.

How to Design a Slide Valve.—The method of designing a valve as here given is so simple that it should present no difficulty, and the engineer who learns and *understands* the method will have no trouble in designing and setting a valve.

Although the dimensions of a valve may be worked out by mathematics, it is essentially a drawing board problem, and is better solved in that way. In solving the problem graphically, use is made of a valve diagram for obtaining the lap, angular advance, etc. There are a number of these diagrams of which some writers employ the one devised by Zeuner. The author considers the Bilgram by far the best and simplest and it is the one here used. The Zeuner diagram is objectionable in that it is a "cut and try" method.*

The following example will serve to illustrate the application of the Bilgram diagram.

Example.—A 7×7 engine is to be run at 450 revolutions per minute. What are the principal dimensions of the slide valve and ports for a steam velocity of 8,000 feet per minute through the port opening and 6,000 feet through the ports? Lead 1/16 inch, cut off ¾, release .9 stroke, length of ports .8 the diameter of the cylinder, and length of connecting rod 2½ times the stroke.

1. Find area and dimensions of port opening and port.

$$\textit{area port opening in sq. ins.} = \frac{\textit{area piston in sq. ins.} \times \textit{piston speed in feet}}{8{,}000}$$

$$= \frac{(7^2 \times .7854) \times (450 \times {}^7/_{12} \times 2)}{8{,}000} = 2.53 \text{ sq. in.}$$

Since the velocity of the steam through the port is reduced to 6,000 feet per minute, it is made larger than the port opening in the proportion of 8,000 ÷ 6,000, or

$$\textit{area of port} = 2.53 \times \frac{8}{6} = 3.37 \text{ sq. in.}$$

---

*NOTE.—Mr. Halsey in his admirable book on Slide Valve Gears, says, in criticising the Zeuner diagram: "The leading data that are given in designing a valve motion are the point of cut off, the port opening, and the lead of the valve (not the lead angle of the crank, as is often conveniently assumed). It is the radical defect of the Zeuner diagram that none of these dimensions can be laid off from known points. The lead must be laid off from an unknown point of the center line, and the port opening from an unknown point on an unknown line. Finally, through these unknown points and the center of the shaft the valve circle is to be drawn from an unknown center and with an unknown radius. Under these circumstances the result sought is found only through blind trial." Continuing he says: "With Mr. Bilgram's method all this is changed. The lead is laid off from a fixed line, the port opening from a fixed point, and the cut off position of the crank is located. The lap circle is then drawn tangent to these lines, and the problem is solved. Moreover, the awkward conception of the backward rotation of the crank is obviated. Finally, these marked advantages are not accompanied by any compensating disadvantages whatever." The author is in accord with the above views.

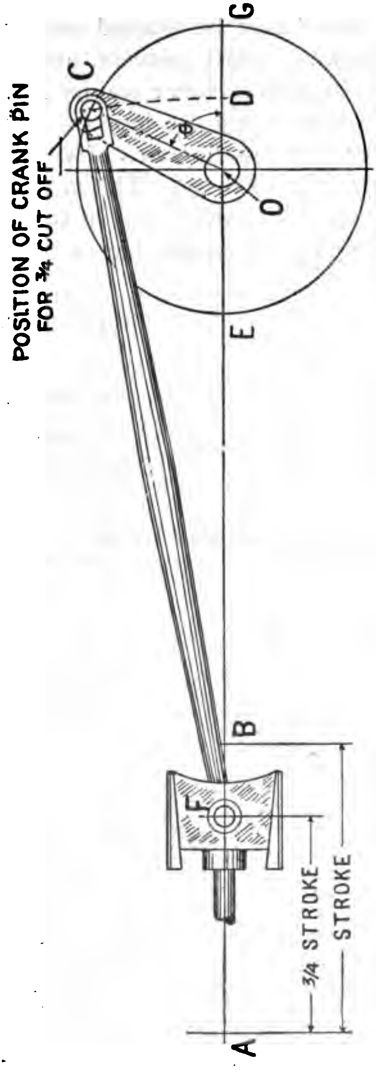


FIG. 387.—Diagram for finding the crank position corresponding to a given piston position. Since the piston and cross head move in unison, the latter is used. With the wrist pin as center, and connecting rod as radius, an arc D C, is described cutting the crank pin circle in C. A line connecting the center of the circle with C, will give the corresponding crank position.

The length of the ports is made .8 the cylinder diameter, or

Length of ports = 7 × .8 = 5.6, say 5½ ins.

Width of steam ports

= area ÷ length

or

3.37 ÷ 5½ = .612, say ⅝ in.

Width of port opening =

$$\frac{5}{8}\times\frac{6,000}{8,000}=\frac{15}{32}\text{, say }\frac{1}{2}\text{ in.}$$

2. Find the position of the crank for ¾ cut off.

In fig. 387 draw a horizontal line, and near one end describe a circle with any radius as O E. Measure off the length of the connecting rod A E = 5 × O E, and A B = 2 × O E. A B, then is the length of the stroke.

Find the position of the cross head F, when the piston is at ¾ stroke by taking A F = ¾ A B. With F, as center and radius F D = A E, describe an arc cutting the circle in C. This is the position of the crank pin when the piston is at ¾ stroke.

3. Find the lap, linear advance and travel of the valve.

Use is here made of the ***Bilgram diagram,*** and the successive steps in its application are as follows:

a. A horizontal line K N, is drawn, and the crank position for ¾ cut off transferred from fig. 387 to fig. 388.

b. Draw a line M S parallel to K N at a distance above ($\frac{1}{16}$ in.) equal to the lead; M S, then is the lead line.

c. With a radius O A, equal to the port opening (½ in.), describe the port opening circle.

d. Now, find by trial the radius E F, and center E, of a circle that shall be tangent to the lead line M S, the port opening circle, and the cut off line O C. *The radius* E F, *of this circle is the outside lap.*

e. Draw E H perpendicular to K N, then *the distance* E H *is the linear advance.*

f. Now by the method of fig. 387, find the crank position for release at .9 stroke. In fig. 388, draw this crank position O C′, and a circle tangent to it with center E G. *The radius* E G, *of this circle is the inside lap.*

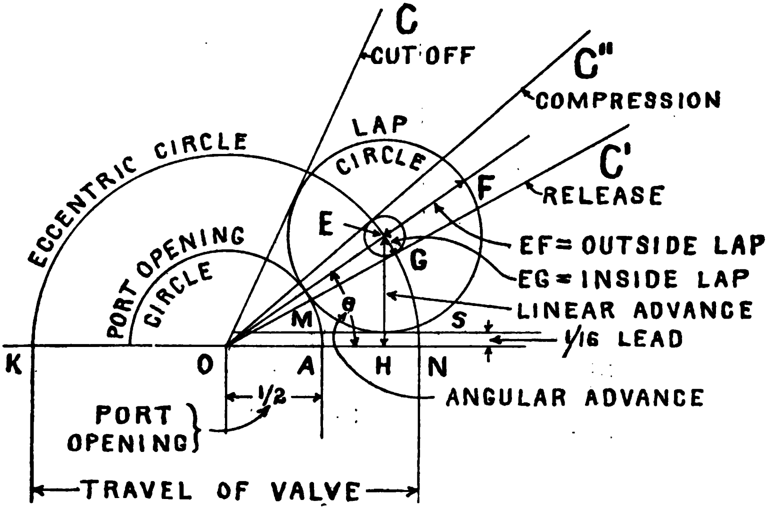


FIG. 388.—The ***Bilgram diagram*** for finding the lap, angular advance travel, etc., of the slide valve. With this diagram, any valve problem may be easily and quickly solved.

g. A second tangent O C″, to this circle gives the crank position for compression.

Measuring the diagram, the dimensions for the valve are

Outside lap = ½ *in.*
Linear advance = 9/16 *in.*
Inside lap = 3/32 *in.*
Travel of valve = 2 *ins.*

With the dimensions just obtained and the given data, the valve and ports may be laid down in the following manner:

1. Find length of valve face.

Length of valve face = outside lap + width of steam port + inside lap $= \frac{1}{2} + \frac{5}{8} + \frac{3}{32} = 1\frac{7}{32}$

In fig. 389 the steam port and one end of the valve is drawn in neutral position giving length of valve face.

2. Find width of exhaust port.

This is done as shown in fig. 390. Draw a horizontal line representing the valve seat and lay off the steam port F G. = ⅝ in. Next lay off the bridge G H* making it ½ inch wide.

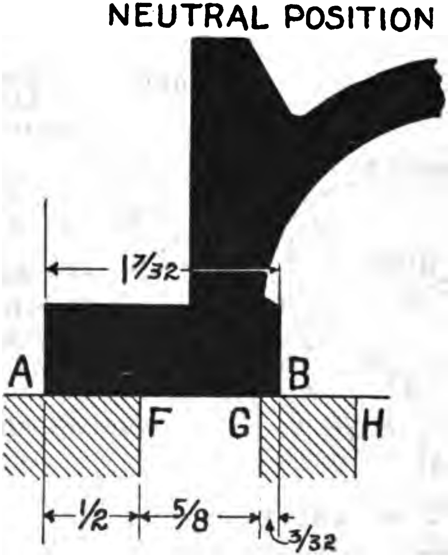


FIG. 389.—***How to lay out the slide valve.*** I. With the dimensions obtained by the Bilgram diagram the length of the face is determined by sketching one end of the valve in its central or ***neutral*** position as here shown.

Draw one end of the valve in its extreme position for admission. For this position, the distance F A, is equal to the *port opening*.

---

*NOTE.—The width of the bridge depends on the size and thickness of the cylinder casting: it should, of course, be amply wide to give a steam tight joint when covered by the valve face.

As steam is being exhausted from the other end of the cylinder when the valve is in this position, it is evident that the exhaust opening B H′, must equal the width of the steam port so as not to choke the exhaust. Hence, lay off B H′ = F G, and draw the bridge H′ G′ = G H. H H′ then is the required width for the exhaust port.

3. Locate the seat limit.

Draw one end of the valve in its extreme position for exhaust as shown in fig. 391. To do this, lay off G B′, equal to the exhaust lap and draw the dotted line which is the neutral position of the exhaust edge of the valve. For the extreme position this edge moves to the left a distance B′ B, equal to one-half the travel of the valve (= 1 in., see page 213).

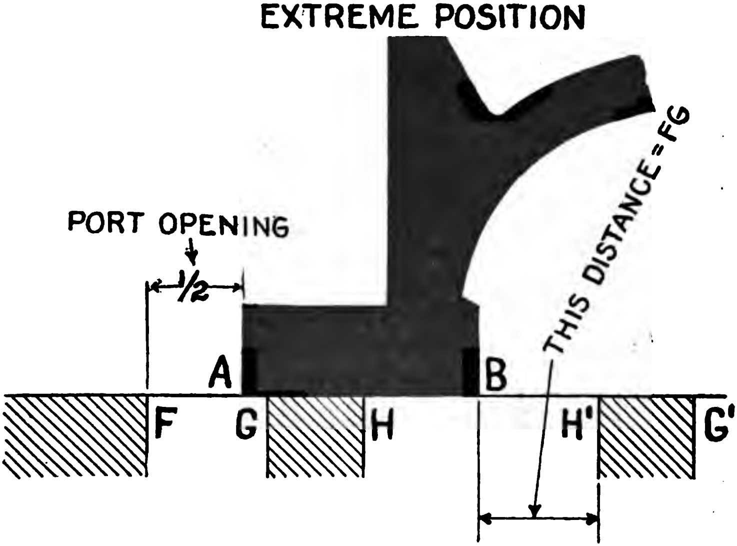


Fig. 390.—***How to lay out the slide valve.*** II. The width of the exhaust port is obtained by sketching one end of the valve in extreme position for admission. Evidently H′, must be so located that B H′ = F G, in order not to choke the exhaust.

Valve end is now drawn in its extreme position thus found, and the seat limit E, (fig. 391) may be located at any point between A and B, which gives sufficient *seal* E B, to prevent leakage of the steam and a clearance A E, for over travel. In this case the over travel is taken at one-half inch. It is recommended for unbalanced valves that the seal E B, be made no more than is necessary for a steam tight joint to reduce the unbalancing.

4. Draw valve seat and valve in neutral position.

From the dimensions already obtained the valve seat is laid down as shown in fig. 392. The two faces of the valve A B and C D, are located in neutral position and the remainder of the valve drawn.

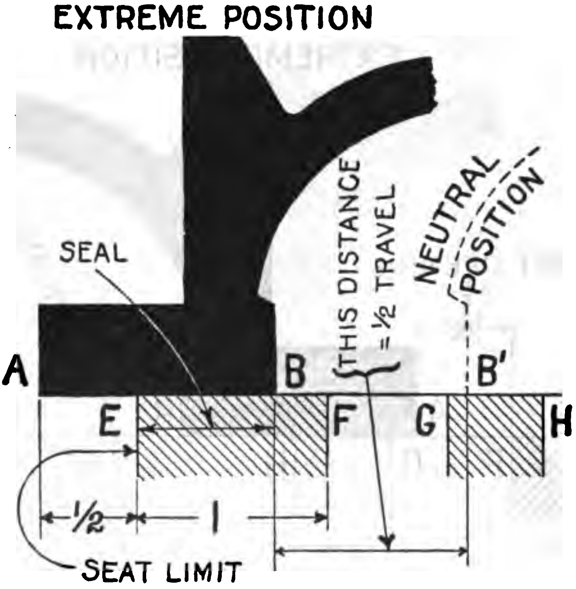


Fig. 391.—*How to lay out the slide valve.* III. The seat limit is obtained by sketching one end of the valve in extreme position for exhaust as shown and locating the seat limit E between A B, at some point which will give sufficient contact E B, for a tight joint, this distance being called the *seal.*

The dimension now needed is the distance B C, between the exhaust edges. After measuring this distance it should be checked as follows:

Distance between exhaust edges = width of exhaust port + 2 × width of bridge — 2 × exhaust lap. That is

$$B\,C = H\,H' + 2\,G\,H - 2\,G\,B$$
$$= 1\tfrac{7}{32} + 2 \times \tfrac{1}{2} - 2\tfrac{3}{32} = 2\tfrac{1}{32}.$$

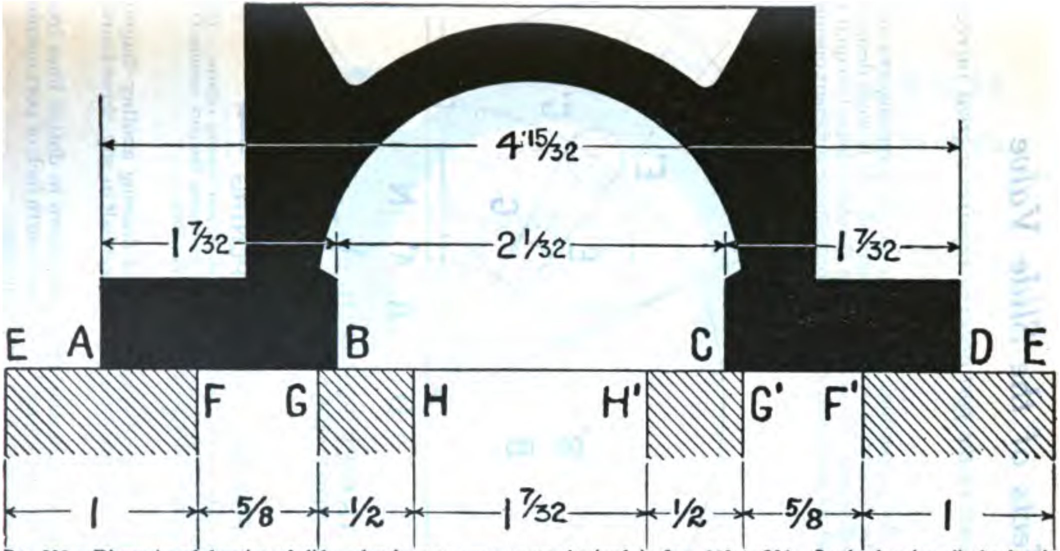


FIG. 392.—Dimensioned drawing of slide valve from measurements obtained in figs. 389 to 391. In the drawing all the longitudinal dimensions necessary for proportioning the steam features are given. The thickness of valve walls will depend on the type of engine, steam pressure, etc.

NOTE.—***In general,*** the valve should be made as light as possible consistent with adequate strength to withstand the steam pressure, so as to reduce its inertia, especially in high speed engines and also to reduce the load on the valve stem in the case of vertical engines.

NOTE.—***The exhaust cavity*** should be made of very liberal height when conditions permit; in special cases its minimum height is equal to the width of the steam port; if less, the exhaust area would be reduced and the exhaust choked. Since the direction of the steam flow is reversed in exhausting, extra resistance is introduced, which increases the back pressure, hence at this point (*i. e.* the valve cavity) extra area should be provided so as to reduce the velocity of the steam.

NOTE.—***In long exhaust lines,*** a pipe of one or two sizes larger than the engine outlet should be used, as the increased efficiency due to the reduction in back pressure thus secured, will more than offset the extra cost of pipe.

Defects of the Slide Valve

1. *A small increase in the port opening requires a great increase in lap and travel.*

A valve having the proportions of the one in the foregoing example, would be suitable for an engine with a fixed cut off. If it were designed for a variable cut off engine, the port opening should at least be equal to the width of the port, or somewhat greater to provide for sufficient opening at early cut off. The effect of this is seen by drawing another diagram as in fig. 393, increasing the port opening from one-half to, say, three-fourths inch.

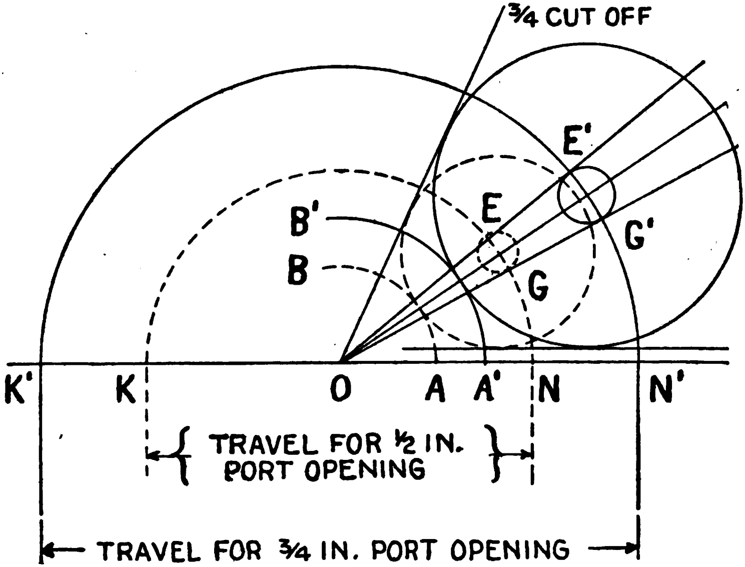


FIG. 393.—*Defects of the slide valve: A small increase in the port opening requires a large increase in the travel, and additional lap.* In the diagram increasing the port opening O A, to O A′, ¼ inch, increases the travel K N, to K′ N′, or 1⅛inches.

In the diagram, the original proportions are shown in dotted lines from which it is seen that for an increase A A′, of one-fourth inch of port opening the travel K N, is increased to K′ N′, or 1⅛ inches.

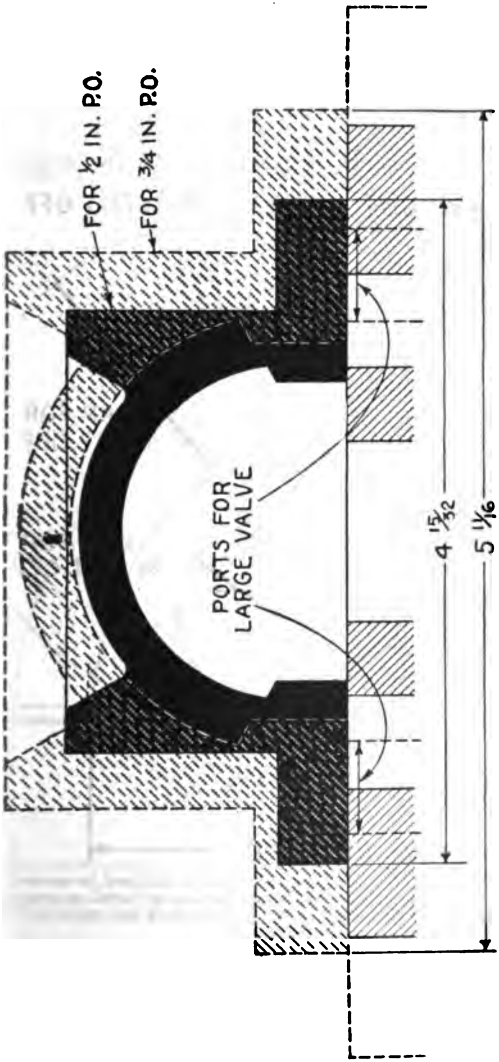


Fig. 394.—Comparison of the two valves corresponding to the diagrams fig. 393, showing the large increase in size of valve necessary for a small increase in port opening. In the figure the valve in full section is for ½ inch port opening, and the one in dotted lines, for ¾ inch port opening.

In fig. 394 the original valve is shown in cross section and one having the new proportions in dotted lines. Comparing the two, it is seen that the exhaust cavity is larger. This together with the additional lap, considerably increases the length of the valve, thus exposing a larger unbalanced area to the action of the steam.

Quite as objectionable is the increase in the travel, requiring as it does, larger ports for moving the valve.

2. *The travel is excessive when the valve is designed for an early cut off; considerable lap is required.*

The effect of shortening the cut off from three-fourths to, say, one-half stroke is shown in fig. 395. The full lines show the proportions for three-fourths cut off, and the dotted lines for one-half cut off. By measuring the diagram it is found that the travel has increased from 2 ins. to $3\frac{17}{32}$ ins., also the lap has increased from ½ to 1¼.

The size of the valve is greatly increased as shown in fig. 177. The three-fourths cut off valve is shown in cross section and the one-half cut off valve in dotted lines. The valve seat for the latter being shown in dotted cross section.

3. *On account of the large proportions and excessive travel necessary, the slide valve is considered undesirable for cut offs shorter than one-half stroke.**

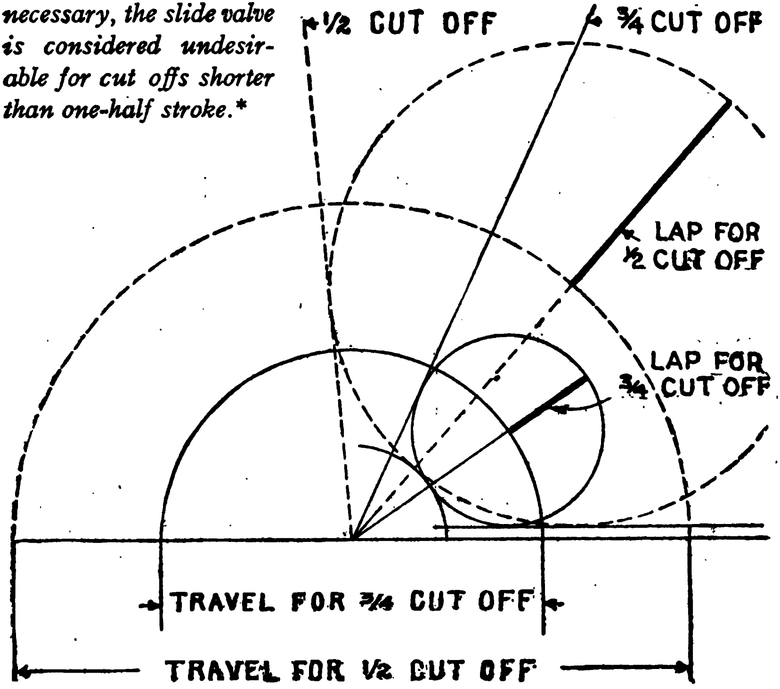


FIG. 395.—*Defects of slide valve: The travel is excessive when the valve is designed for an early cut off; considerable lap is required.* The diagram shows the large increase in travel necessary in shortening the cut off from ¾ to ½ stroke, and also why the slide valve is not suitable for cut offs shorter than ½ stroke.

In general, to keep the proportions of a valve within limits, for a short cut off, the ports must be made as long as possible so as to reduce the width and thus secure the proper admission with a minimum of valve movement.

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*NOTE.—It should be understood that this means the cut off with full travel. Any valve may be made to cut off shorter by reducing the travel as before explained but this is at the expense of reducing the port opening.

The Allen valve requires only one-half the lap and travel of the ordinary valve as shown in the diagram fig. 396*, the diagram for the Allen valve being shown in full lines, the other dotted. In figs. 397 and 398 are shown the proportions required for the ordinary valve cutting off at three-fourths and one-half, and the Allen valve with one-half cut off. M, is the half travel required for three-fourths cut off, M′, for one-half cut off, and M″, for one-half cut off with the Allen valve. The figures are intended to show by comparison the undesirable features of the ordinary valve at early cut off.

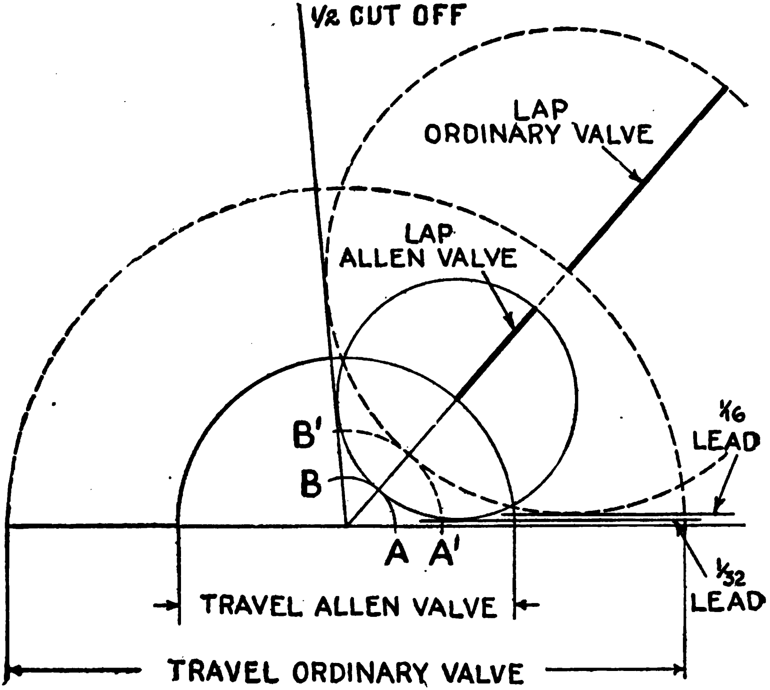

FIG. 396.—Comparative diagrams showing travel of Allen and plain slide valve. The Allen valve requires only half the travel and half the lead of the plain slide valve.

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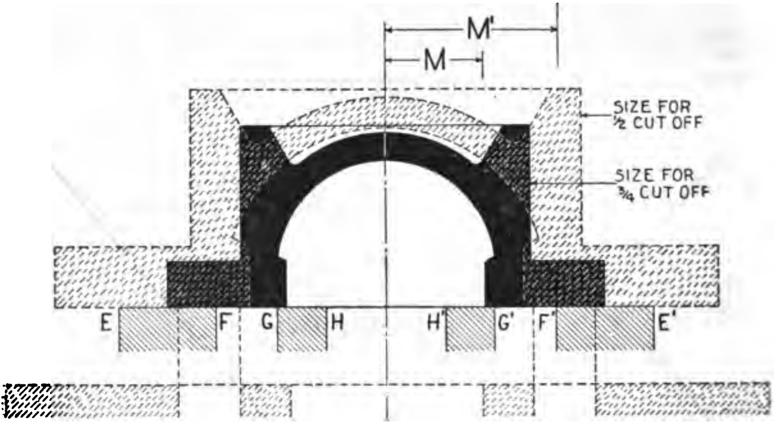
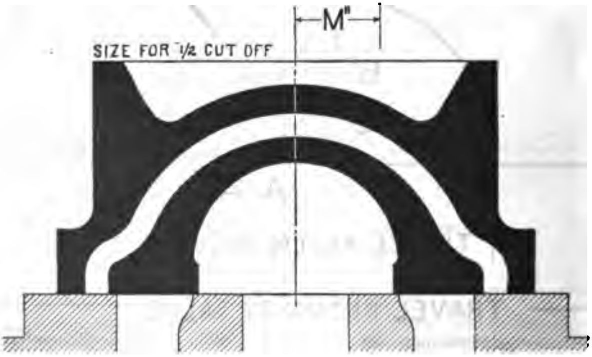
*NOTE.—In the diagram the port opening arc A B, is taken with radius one-half smaller than A′ B′, the port opening arc of the ordinary valve because *double admission* is secured with the Allen valve.

The full benefit to be derived from the Allen valve may be obtained by so modifying the steam ports, as shown in figs. 383 and 398, that the supplementary ports are not closed when the valve is in the extreme positions.





Figs. 397 and 398.—Showing sizes of valve for ½ and ¾ cut off corresponding to the diagram fig. 396. The considerable increase in the length of valve and seat necessary for the shorter cut off as indicated by the dotted lines should be noted. Fig. 398 shows the Allen valve with modified steam ports for ½ cut off as compared with the plain slide valve for ½ and ¾ cut off.

4. *For a short cut off, release and compression occur too early.*

This is illustrated in the diagram fig. 399. A, A', and B, B', are the crank positions of release and compression for three-fourths and one-half cut off respectively. Either may be corrected by the addition of positive or negative inside lap, but it must be evident that to correct one, will cause the other to occur still more prematurely.

5. *For variable expansion, the port opening is inadequate, and release and compression occur too early at short cut off.*

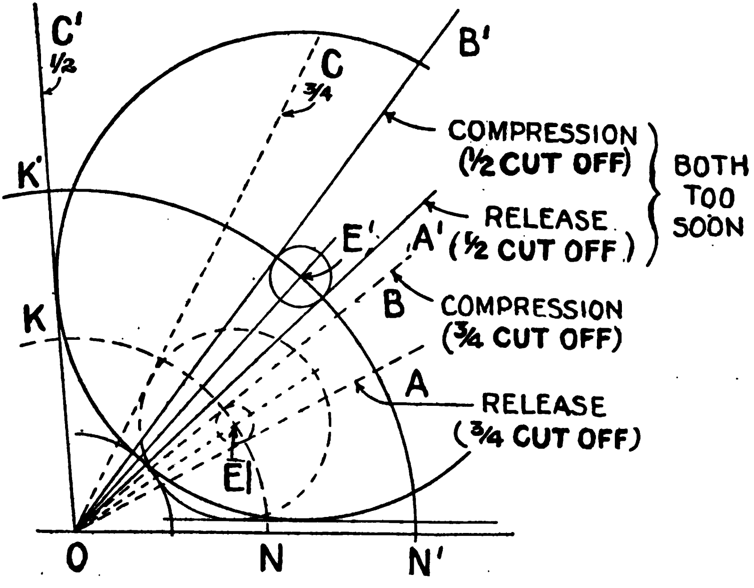


FIG. 399.—***Defects of the slide valve:*** *For a short cut off release and compression occur too early.* The diagram shows the effect on release and compression, of changing the cut off from ¾ to ½. Either may be corrected with positive or negative inside lap but the error of the other will be correspondingly magnified.

As will be explained in detail in a later chapter, the cut off may be varied by the method of *combined variable angular advance and variable throw,* as is done mainly on engines having shifting or swinging eccentrics known popularly as "automatic cut off engines."

Assume latest cut off at ¾ stroke and that TT, is greatest travel mechanically feasible. Draw O C = ¾ cut off and describe travel circle with radius O T, and on this circle draw lap circle tangent to O C, and lead line. The corresponding port opening A′ B′, as seen, is considerably in excess of the required or normal opening A B.

Now since the travel is reduced as the cut off is shortened a series of lap circles R, R′, R″, will appear in the diagram on a line M S, drawn parallel to the lead line (assuming *constant* lead). Thus lap circle R′, for normal port opening, gives a cut off O C′, near one half stroke.

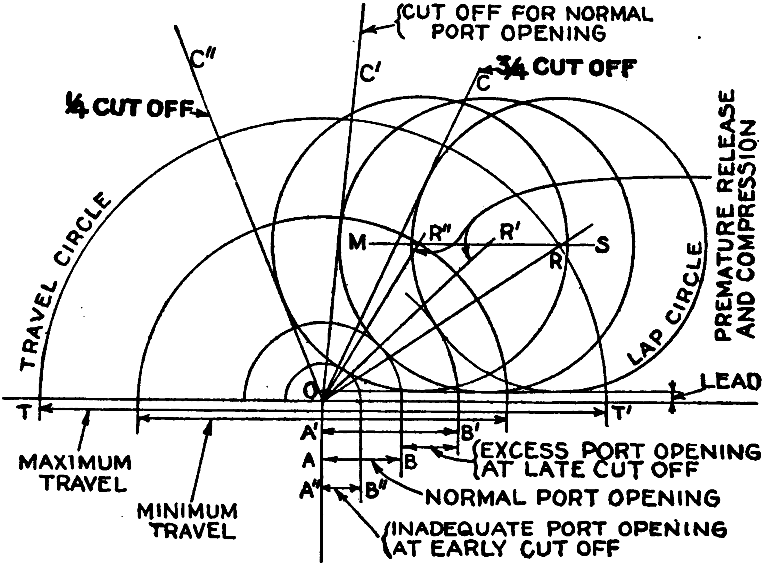

CUT OFF FOR NORMAL PORT OPENING
C″
C′
¾ CUT OFF
C
¼ CUT OFF
PREMATURE RELEASE AND COMPRESSION
TRAVEL CIRCLE
M
R″
R′
S
R
LAP CIRCLE
LEAD
O
T
T′
A′
B′
EXCESS PORT OPENING AT LATE CUT OFF
MAXIMUM TRAVEL
A
B
A″
B″
NORMAL PORT OPENING
MINIMUM TRAVEL
INADEQUATE PORT OPENING AT EARLY CUT OFF

FIG. 400.—***Defects of the slide valve 5.***—For variable expansion the port opening is inadequate, and release and compression occur too early at short cut off. The diagram shows the effects of changing the cut off from ¾ to ¼ by the method of *combined variable angular advance and variable travel*. Evidently, when variable expansion is thus obtained, the travel of the valve should be as great as possible in "full gear," because of the decreasing port opening with shortening of the cut off. From the diagram, it is also clear that the shorter the full gear cut off, the larger the port opening at earliest cut off.

Ordinarily the most economical cut off is at ¼ stroke (non-condensing), the lap circle R″, in the diagram giving this cut off. The port opening as seen for ¼ cut off is reduced to A″ B″, being about one half less than the required amount A B.

Accordingly the operation of automatic cut off engines at early cut off is characterized by a sloping admission line on the indicator card the pressure drop reducing the efficiency. Moreover release and compression occur as the cut off is shortened, being entirely too early at ¼ cut off. In the diagram these events are represented by the crank position O R, O R′ and O R″, corresponding to cut offs O C, O C′, and O C″ respectively.

CHAPTER 5

THE VALVE GEAR

Ques. What is the valve gear?

Ans. The mechanism, or combination of parts by which a reciprocating, or to and fro motion is imparted to the valve from the rotary motion of the shaft. The simplest form of valve gear consists of:

1. Yoke;
2. Stem;
3. Guide;
4. Eccentric rod;
5. Eccentric strap;
6. Eccentric.

These parts are shown assembled in fig. 401.

The Valve Yoke.—The valve is held in position by a yoke which consists of a rectangular band (fig. 402) which surrounds the upper part of the valve. The latter is indicated by dotted lines illustrating the position of the valve with respect to the yoke.

The fit between the yoke and valve is such that the latter is free to move up or down and thus adjust itself to the seat. At A, is a slight enlargement to receive the valve stem.

In some cases the yoke is omitted and the stem attached direct to the valve as shown in fig. 403.

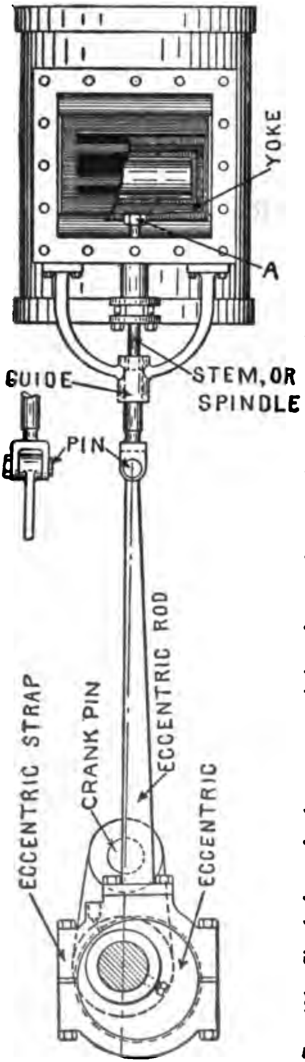


FIG. 401.—Simple form of valve gear, consisting of eccentric, eccentric strap, eccentric rod, pin, guide, stem or spindle with adjustable end, and yoke.

The valve is retained in place by the nuts A and B, by which the length of the stem is adjusted in setting the valve. In fig. 402, the adjustment of the stem is at A.

The Valve Stem.—This is usually made of steel and serves to transmit the motion of the eccentric rod to the valve. Fig. 402 shows the type stem used with a yoke, and fig. 403 the form used without yoke. From figs. 404 to 406 it is seen that a valve stem consists essentially of:

1. A threaded section A forming a connection for yoke or valve, and providing for adjusting the length of the stem in setting the valve;

2. A cylindrical section B, which passes out of the steam chest through the stuffing box;

3. An enlarged section C to secure sufficient bearing area for the guide;

4. A pin connection D for the eccentric rod.

NOTE.—The valve stem is designed to move the valve under the most unfavorable conditions. A short rule is: diam. stem $= \frac{1}{3}$ diam. cylinder $= \frac{1}{8}$ diam. piston rod. Seaton's formula: $d = \sqrt{\frac{p\,L\,B}{F}}$, where p = boiler pressure; L and B, length and breadth of (slide) valve; F = 10,000, for long *iron* stem = 12,000 for long *steel* stem.

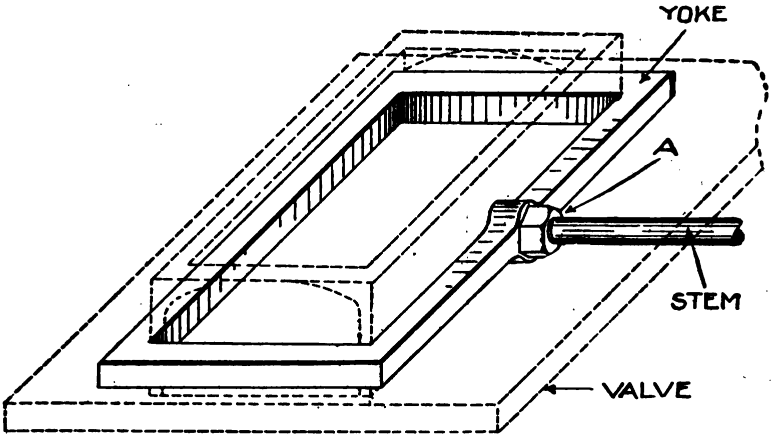


Fig. 402.—The valve yoke, or rectangular frame which embraces the box shaped section of the valve, and to which the valve stem is attached. The valve is shown in dotted lines to indicate the position of the yoke.

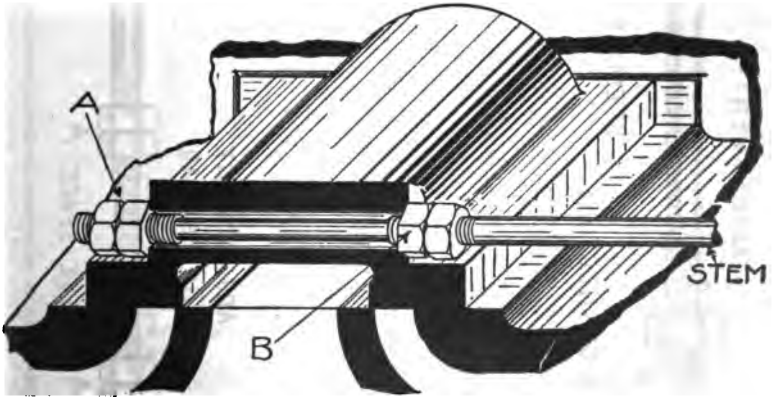
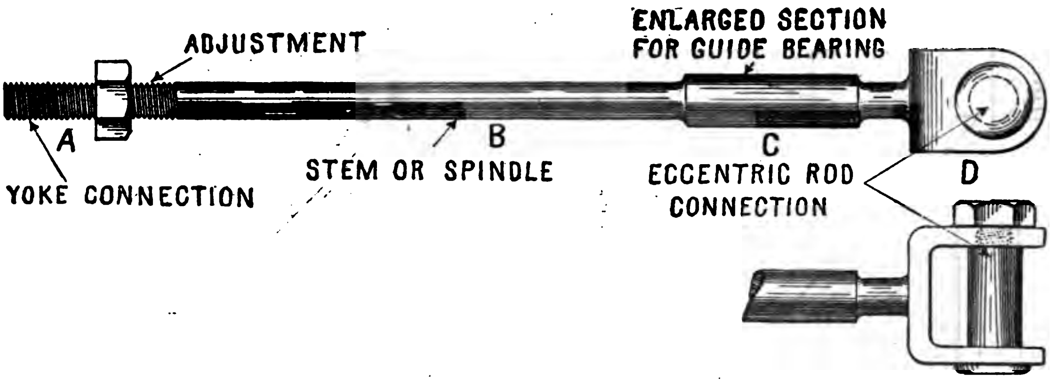


Fig. 403.—Method of connecting the valve stem to the valve without a yoke. The stem passes through a circular section cast in the valve, being adjustable by means of the nuts A, B, at each end.



FIGS. 404 and 405.—Form of valve stem for yoke connection. The length of thread at A, is made sufficient for adjustment. At C, the stem is enlarged for the guide bearing, there being a forked end which carries the eccentric rod pin.

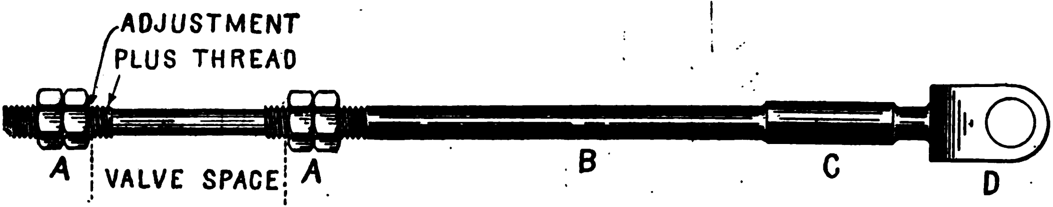


FIG. 406.—Form of valve stem for direct connection without yoke. There are two pairs of adjustment nuts A, A; the diameter of the rod between the threaded sections is reduced to make a plus thread at the end.

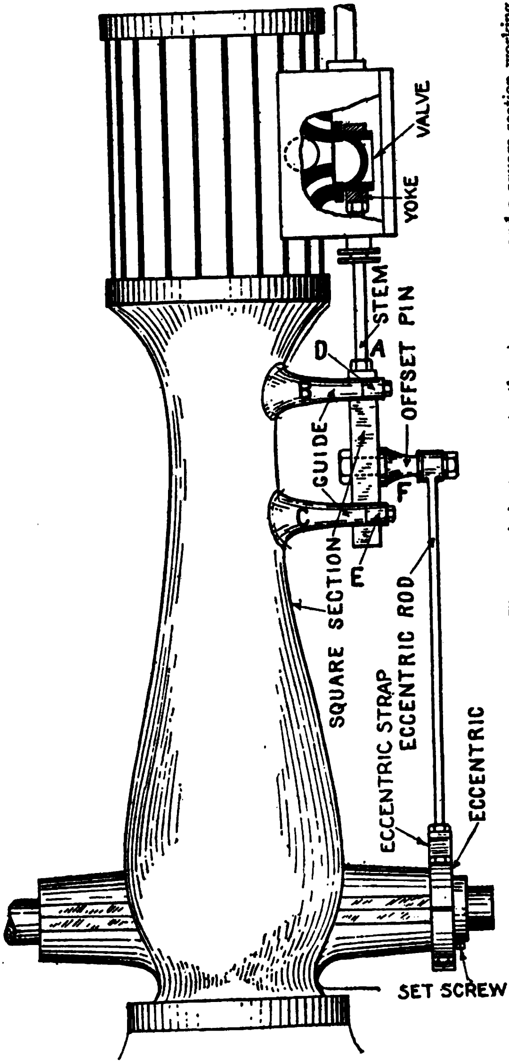


FIG. 407.—Simple valve gear with offset eccentric. The stem is in two parts, the stem proper, and a square section working through guides, and to which is attached an offset pin.

In many cases the eccentric cannot be placed directly in line with the valve stem on account of the room required for the main bearing. To provide for this the eccentric rod connection is placed between the valve and the guide, and an *offset pin* is attached to the enlarged section projecting out to the line of the eccentric rod as shown in fig. 407. Here, it is necessary to make the enlarged section of the valve stem square or rectangular, to prevent the stem turning due to the angular thrust of the eccentric rod.* On account of this turning action, *it is important that there be no lost motion in the guide bearings.*

The Guide.—

The object of the guide is to

---

*NOTE.—In this type of valve stem, the square section is usually a separate part with a threaded joint A, (figs. 402 and 407), to facilitate construction and adjustment.

prevent the previously mentioned turning motion, and any side movement of the valve stem which may occur on account of the action of the eccentric rod. In fig. 408, the guide consists of a U shaped piece attached to the lower end of the steam chest. This style is used extensively on marine engines when the eccentric is directly under the stem; it is, however, somewhat in the way when packing the stuffing box.

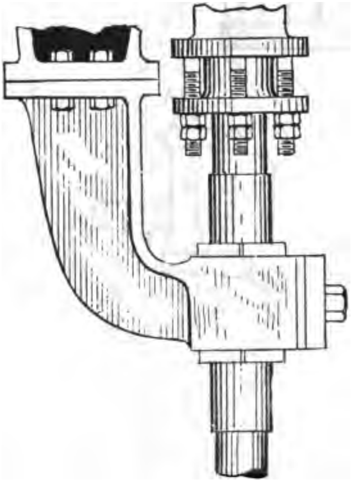


Fig. 408.—Detail of marine type of valve stem guide with adjustable brasses.

The valve stem guide shown in fig. 401 has no means of adjustment for wear. A more desirable form is shown in fig. 408 having a box with adjustable brasses; this is the regular marine type as used on large engines.

On engines having offset eccentrics, there are usually two guides which are attached to the engine frame as shown in fig. 407.

Two projecting arms B, C, are cast to the frame with cross pieces D, E, bolted on, and serving as guides. The offset pin F is screwed into the square section of the stem and locked by a nut.

There are numerous forms of guide depending on the design of the engine. In fig. 409 is shown the guides of the Brownell engine. Lubricating devices are attached to the guides and below is a tray to catch the waste oil.

Rocker Levers.—Sometimes an offset pin is carried on two rocker levers, M, and S, as shown in fig. 410, which illustrates the construction used on the American Ball engine.

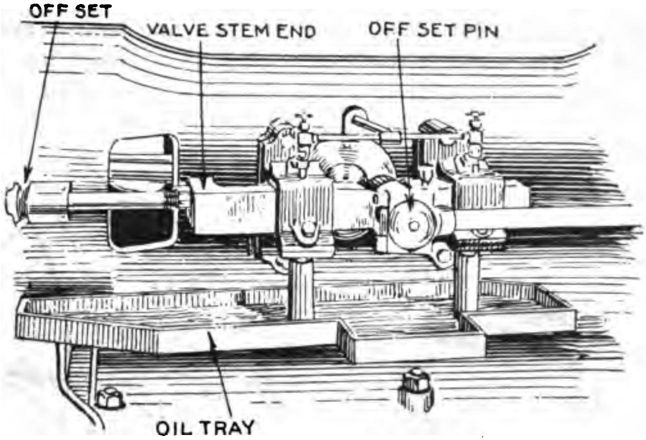


FIG. 409.—Valve stem guides, and offset pin of the Brownell engine. The figure shows also the oiling devices, and a drain tray for the oil.

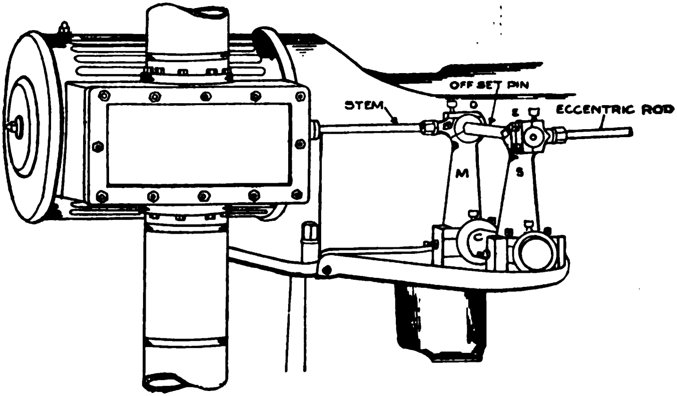


FIG. 410.—The American-Ball engine. View showing rock shaft C, and rocker levers M, S, which carry the offset pin, the latter being attached at D and E.

The pin is carried by two *rocker levers* M, and S, keyed to the shaft C. There are pivoted joints at D, and E, for the valve stem and eccentric rod.

In operation, the rockers move through a small arc, and a slight up and down movement is produced owing to the circular path of D. The stem is sufficiently flexible to allow for this, hence no special provision is made. This style of offset pin is used to advantage on engines having the eccentric rod located outside the flywheel where the offset, or distance D E, is considerable.

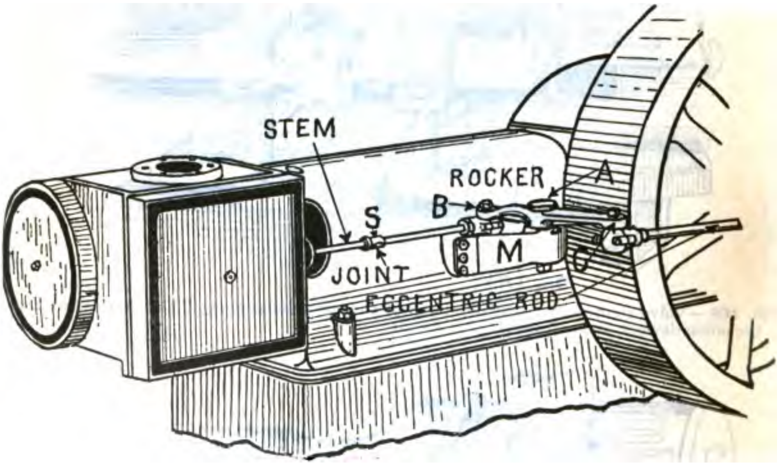


Fig. 411.—Erie City high speed engine. An *indirect* rocker fulcrumed at A, is used in place of an offset pin. The valve stem is attached at B, and the eccentric rod at C. To allow for side motion due to the rocker, a flexible joint is provided at S.

Another type of rocker used on engines of this class consists of a horizontal lever, pivoted near the center as shown in fig. 411.

A projection M, bolted to the engine frame carries the lever whose fulcrum is at A, connection at B, being made with the valve stem, and at C, with the eccentric rod. Since the arm A B, is rather small, the arc described by B, will cause more side motion of the stem than is occasioned by the movement of D, in fig. 410, hence a flexible joint is provided at S, to relieve the stem of undue bending.

Ques. What is a *direct* valve gear?

Ans. One in which the valve stem and eccentric rod move in the same direction as shown in fig. 410.

Ques. What is an *indirect* valve gear?

Ans. One in which the valve stem and eccentric rod move in opposite directions as shown in fig. 411.

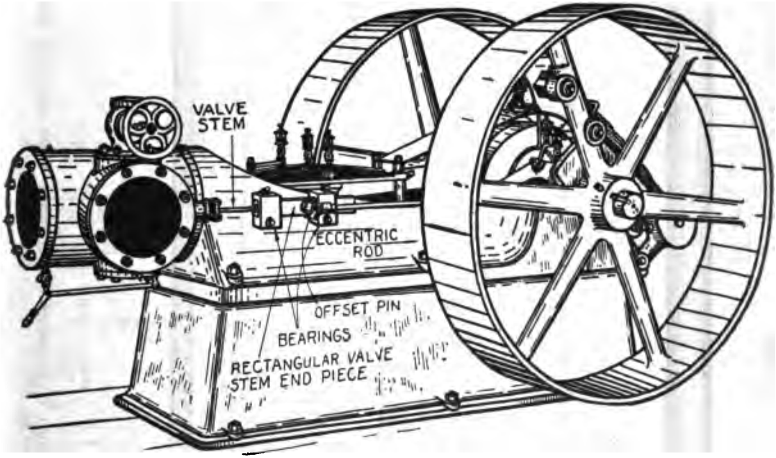


FIG. 412.—"Vim" horizontal automatic cut off engine with *direct* valve gear without rocker. The eccentric rod, as seen, is offset with pin attached to a wide rectangular valve stem end piece working in two bearings to prevent lateral or twisting motion.

The Eccentric Rod.—Motion is transmitted from the eccentric to the valve stem or rocker by the eccentric rod. The rod may be either round or of rectangular section. A simple form of rod is shown in fig. 413 with the end connections indicated by dotted lines. In fig. 415 is shown a rectangular rod which is secured to the strap by a cross piece A, and stud bolts B and C.

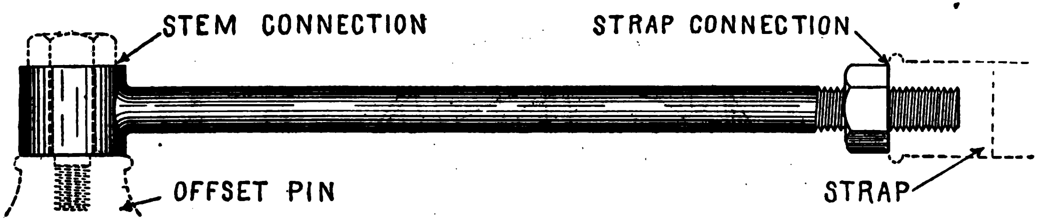
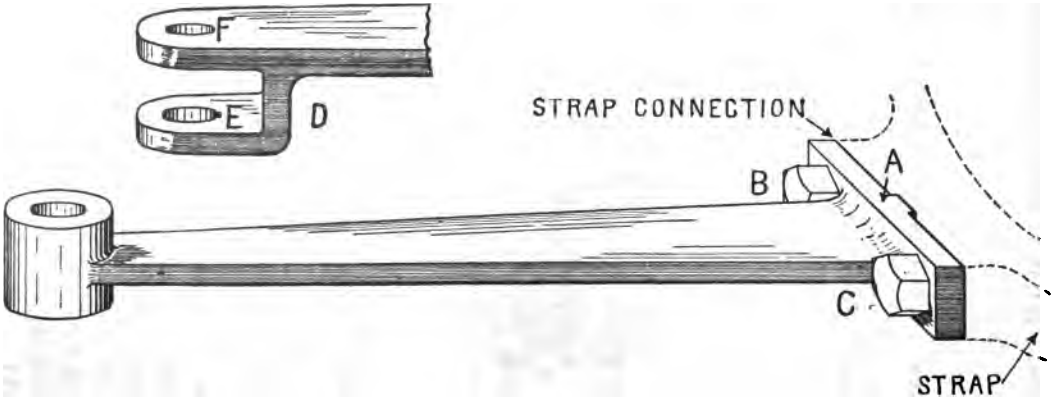


FIG. 413.—Plain eccentric rod having threaded connection for strap, and an eye for the valve stem pin.



FIGS. 414 and 415.—Rectangular rod with T end connection for strap. When used with link motion the other end has an offset fork which carries the pin as shown in fig. 414.

Where two eccentrics are employed, as with link motion, the pin end is offset as shown at D, fig. 414, the offsets of the two rods being on opposite sides to permit the link to work centrally. The pin is held by the two jaws, E and F.

On marine beam engines the eccentric rods are of great length and to make them rigid each rod is usually built up of flat wrought iron bars in shape of a tapering lattice girder. The extreme end is a solid bar, with a notch for hooking on to the rocker pin.

A pin is used in place of an eccentric on many high speed engines where the rod is located outside the fly wheel; the rod being constructed as shown in fig. 416. Part of the fly wheel and the rocker are indicated in dotted lines to show the position of the rod.

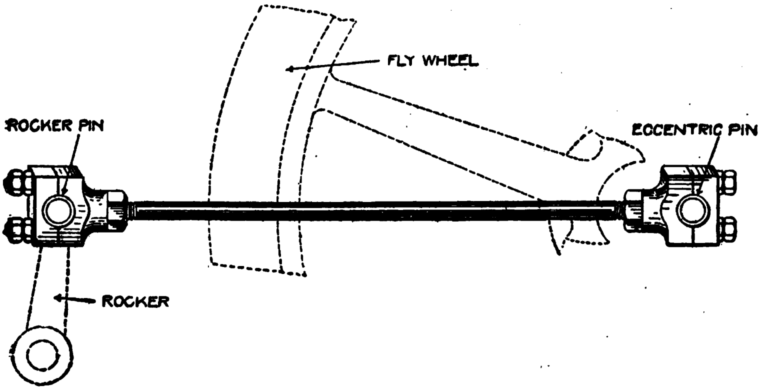
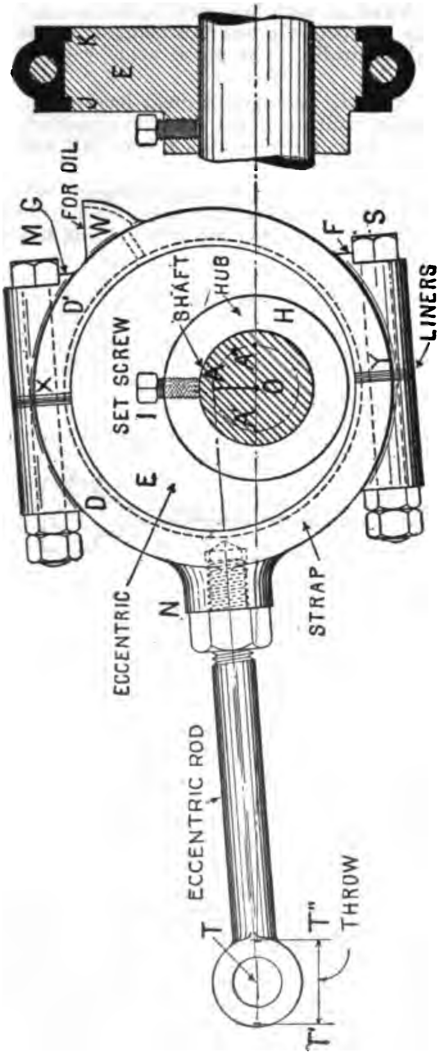


Fig. 416.—Outside eccentric rod of a high speed engine. With rods of this type, an eccentric pin is used in place of an eccentric. The other end of the rod is usually attached to a *direct rocker*. This and part of the fly wheel are shown in dotted lines.

The Eccentric and Strap.—An eccentric is the equivalent of a *crank pin which is so large in diameter that it embraces the shaft to which it is attached and dispenses with arms.* Its object is to change the rotary motion of the shaft into a reciprocating, or to and fro motion. This motion is transmitted by the strap and eccentric rod to the valve stem and valve. An eccentric and strap of simple construction is shown in figs. 417 and 418.

The eccentric E, is a cast iron disc having a projecting boss or hub H, containing a set screw I, to secure it in any position on the shaft. A, is



Figs. 417 and 418.—Eccentric strap, and eccentric rod. The eccentric E, is usually secured on the shaft O, by a set screw I. A groove is turned in the strap to register with a projection on the eccentric so as to prevent side motion. The strap is in two halves, D, D' and held together by the bolts M, S, liners being inserted between for adjustment. O, is the center of the shaft, and A the center of the eccentric. O A, is the *eccentricity*; A' A", or *twice* the eccentricity O A, is the throw. A' A" = T' T", the movement of the valve stem pin.

the center of the eccentric, and O, the center of the shaft. The hole for the shaft is drilled out of center or "eccentric" with the center of the disc, hence the name *eccentric.*

The distance O A between the center of the shaft and the center of the eccentric is the *eccentricity* and is equal to *one half* the throw. *This distance is sometimes wrongly called the throw.*

Throw

Ques. What is the throw of an eccentric?

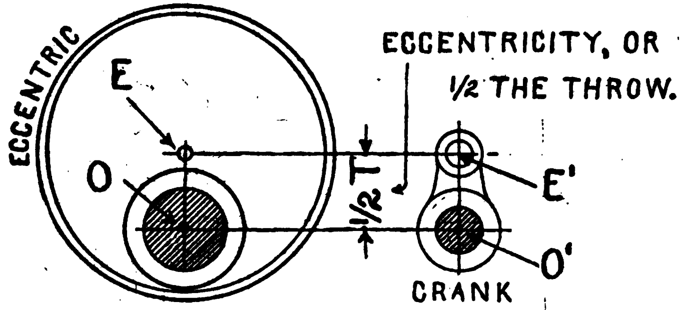
Ans. *Twice* the eccentricity or the amount of to and fro movement produced.

The throw is equal to the diameter of the circle described by the center of the eccentric as it revolves around the shaft. Thus, in figs. 417 and 418, when the center of the eccentric is at A' and the end of the rod at T′, the end of the rod moves from T′ to T″ as the center of the eccentric moves from A′ to A″. *The distance* T T′ *or* A′ A″ *is the throw,* some erroneously call half this distance the throw.

The eccentric is embraced by a strap usually made in two pieces D, D′. These are held together by the bolts M, S, liners being inserted at X and Y for adjustment. The circumference of the eccentric is recessed at J and K, to register with a groove in the strap; this prevents any side motion of the strap.

The eccentric rod is attached to a projection or neck N, usually by a threaded connection as shown. At W, is an oil well for lubrication.

The strap is recessed at F and G, to register with a side of each bolt which prevents the latter turning when the nuts are tightened.



FIGS. 419 and 420.—Comparison of eccentric and crank. An eccentric is equivalent to a small crank whose arm O′ E′ is equal to the distance O E between the center of the shaft, and the center of the eccentric. *This distance is the* **eccentricity**, *or* **one-half** *the throw.* Sometimes erroneously called the throw.

There are numerous forms of eccentric, the one shown in fig. 417 and 418 serves to illustrate the principles and parts; it is such as would be used on a small horizontal engine.

Angular Advance

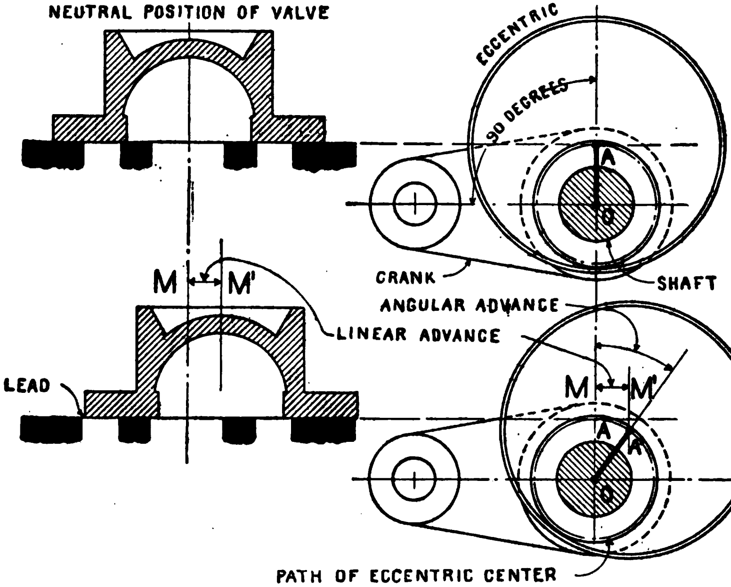
Ques. What is the angular advance of an eccentric?

Ans. The number of degrees the eccentric must be moved forward from a position at right angles to the crank to give the valve its *linear advance*, that is, to move it from its neutral position to its position when the crank is at the beginning of the stroke.

This is illustrated in figs. 421 and 422. In fig. 421, the crank is on the dead center and the valve in its neutral position. The corresponding position of the eccentric is shown 90 degrees ahead of the crank.

When the crank is on the dead center the valve must be in the position shown in fig. 422, a distance M M′, to the right equal to *the lap plus the lead or linear advance.* Hence the eccentric must be turned ahead on the shaft far enough to move the valve this distance from its neutral position.

To find the angular advance, M M′, is measured off to the right of the vertical line and a parallel line drawn. This cuts the path of the eccentric center at A′, from which it is evident that the eccentric must be turned ahead through the arc A A′, to move the valve to the position M′, A O A′ being the *angle of advance,* or *angular advance* as it is called.



FIGS. 421 and 422.—Illustrating *linear,* and *angular advance.* When the crank is on the dead center, and the eccentric set 90° ahead, the valve should be in its neutral position as shown in fig. 421. The valve, however, when the engine is on the dead center, must be at a distance (M M′, fig. 422) from its neutral position equal to the *lap* + *lead* or in its position of *linear advance.* The eccentric then must be turned ahead through an angle A O A′, its angular advance, sufficient to move the valve to its linear advance position M′.

Ques. What objections are there to eccentrics?

Ans. The diameter is large in proportion to the throw. On account of this large diameter, the velocity of rubbing against the strap is considerable as compared with an equivalent crank pin. This causes an increase of friction and tendency to heat which requires closer attention to be given to lubrication and adjustment of the strap.

Sometimes the eccentrics on small engines have straps with only a single adjustment as shown in fig. 423.

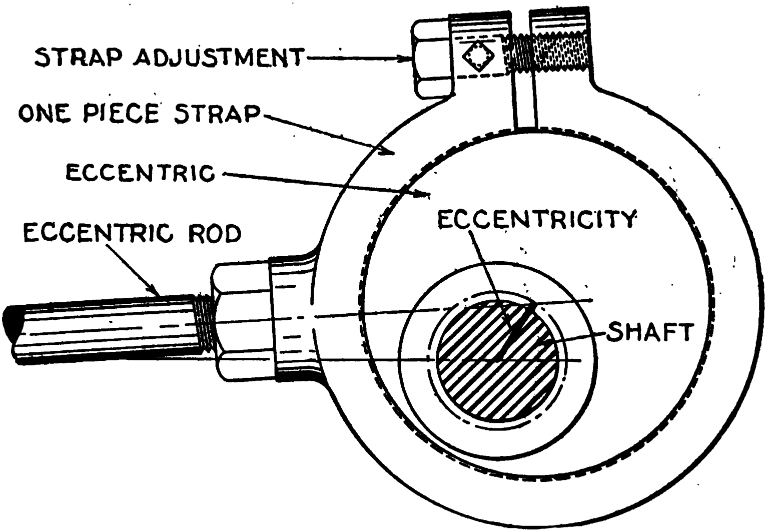


FIG. 423.—Eccentric strap in one piece. A type of strap for use in inaccessible places, as on small multi-cylinder marine engines having cast frames and valves on the side. On account of the poor fit after adjustment, this type of strap is liable to heat, and should be avoided in design wherever possible.

The strap is one piece and consists of a split ring with projecting lugs for the adjusting bolt. Liners may be inserted in the space between the lugs, or a set screw provided to hold the bolt in position.

Eccentrics of this type are regarded by some (including the author) as being only a little better than a makeshift because when wear is taken up the strap looses its circular form and no longer bears properly on the eccentric, making it more liable to heat in operation.

Large eccentrics are usually made in two unequal parts H and H′, as shown in fig. 424.

These are held together by the key bolts M, and S; the keys retain them in position and prevent turning when the nuts are tightened. A keyway K, is provided to retain the eccentric in position on the shaft and also a set screw C. The larger part H, of the eccentric is cast with ribs (R, R′, R″), which reduces the weight and provides room for the bolts and set screw.

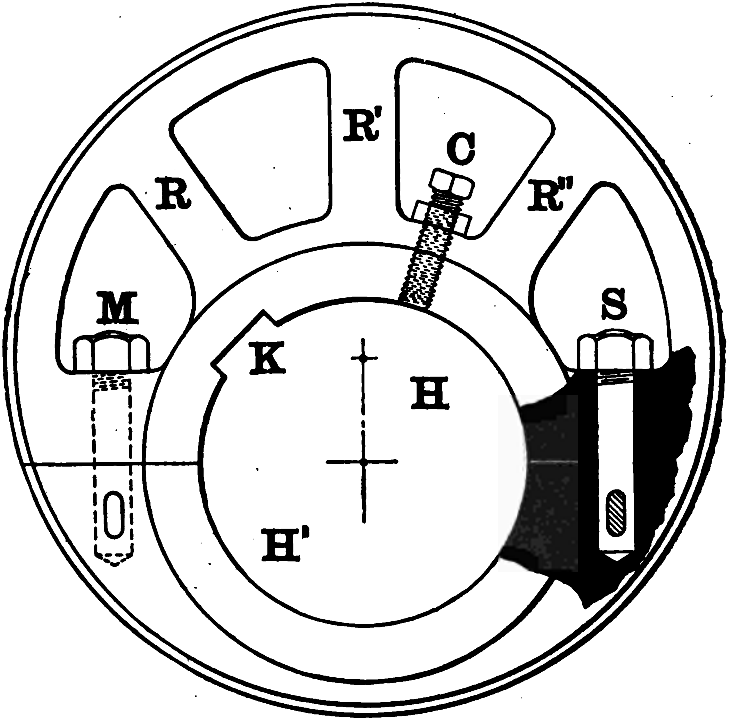
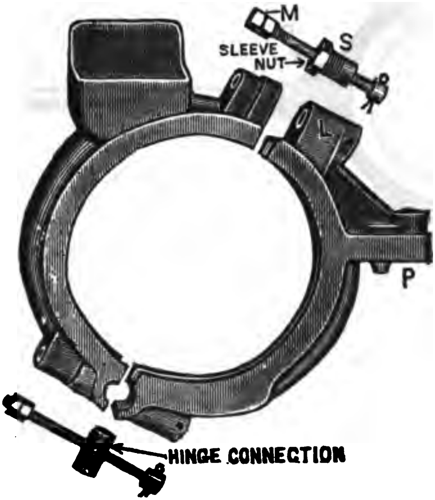


Fig. 424.—Eccentric for large engines. Usually made in two unequal parts H, H′, and held together by the key bolts M, S. The eccentric is retained in position by a key and set screw.

An eccentric strap having micrometer adjustment as used on the Ideal engine is shown in figs. 425 to 428.

The two halves of the strap are joined at the bottom by a construction possessing the elements of a hinge. At the top, the uniting bolt M, passes through a sleeve nut S, which is threaded into the base lug L, and fixes the distance between the two halves. In adjusting the strap, the nuts M, are slacked and the sleeve nut S, turned to regulate the distance between the two halves, and the nuts M, again tightened which holds the two halves rigidly together.



FIGS. 425 to 428.—Eccentric strap of the Ideal engine, having micrometer adjustment. The two halves are joined, at the bottom by a hinge connection, fig. 428, and at the top by a uniting bolt M, fig. 426; this passes through a sleeve nut S, which is threaded into the base lug and fixes the distance between the two halves.

NOTE.—There are several kinds of eccentric: 1, the *circular*, or eccentric properly so called, and 2, the various other contrivances bearing the name of eccentrics, but which are virtually cams, such as the heart shaped eccentric, the triangular eccentric, eccentrics with a uniformly varied motion, etc.

NOTE.—The large amount of friction produced between the eccentric and its strap renders the application of the eccentric impracticable in cases in which it is required to transmit a great force. The same may be said of all contrivances bearing the name of eccentrics; they are applicable only when the force to be transmitted is small.

NOTE.—Because of the relatively high velocity of the rubbing surface as compared with an *eccentric pin*, the latter is the more desirable and is used in best practice where the construction permits, as for instance, in marine engines having valves on the side operated from a valve shaft.

NOTE.—In marine practice, eccentric rods are generally made of steel with bushings at the top ends, and the eccentric straps are generally constructed of cast steel lined with white metal as bearing surfaces, the eccentrics being of cast iron or cast steel, preferably of the former metal except in very light construction.

In fig. 429 is shown the strap and eccentric rod of the Erieco engine. The rod is attached to the strap at A, and held by the set screw D.

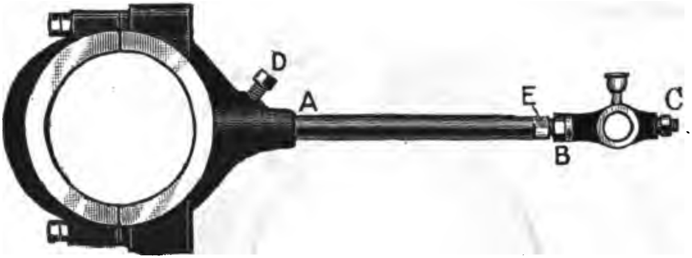


FIG. 429.—Erieco eccentric strap and rod. The strap is lined with Babbitt metal, and the eccentric, being an arc which properly fits the strap, makes a ball and socket joint. This arrangement insures cool running even though there be a lateral error in the alignment.

The length is adjustable by means of the threaded end E, and nut B. Adjustment for the bearing is made at C.

NOTE.—***Eccentric Rod***—The diameter of the eccentric rod in the body and at the eccentric end may be calculated in the same way as that of the connecting rod, the length being taken from center of strap to center of pin. Diameter at the link end = .8 D + .2 in. This is for wrought iron. Eccentric rods are often made of rectangular section.

CHAPTER 6

VARIABLE CUT OFF

Where economy of steam is desirable, engines are used which automatically adjust themselves to any change in the load by altering the cut off. These are called *automatic cut off engines** as distinguished from *throttling engines*, in which the cut off is fixed and the steam supply varied at the throttle valve. In either class the regulation is controlled by an automatic *governor*.

There are several methods employed to vary the cut off as by:

1. Shifting eccentric;
2. Swinging eccentric { axial; offset;
3. Rotating eccentric (independent cut off valve);
4. Fixed eccentric (adjustable lap cut off valve);
5. So called expansion valve gears.†

These various methods may be divided into two classes, according as the cut off is varied:

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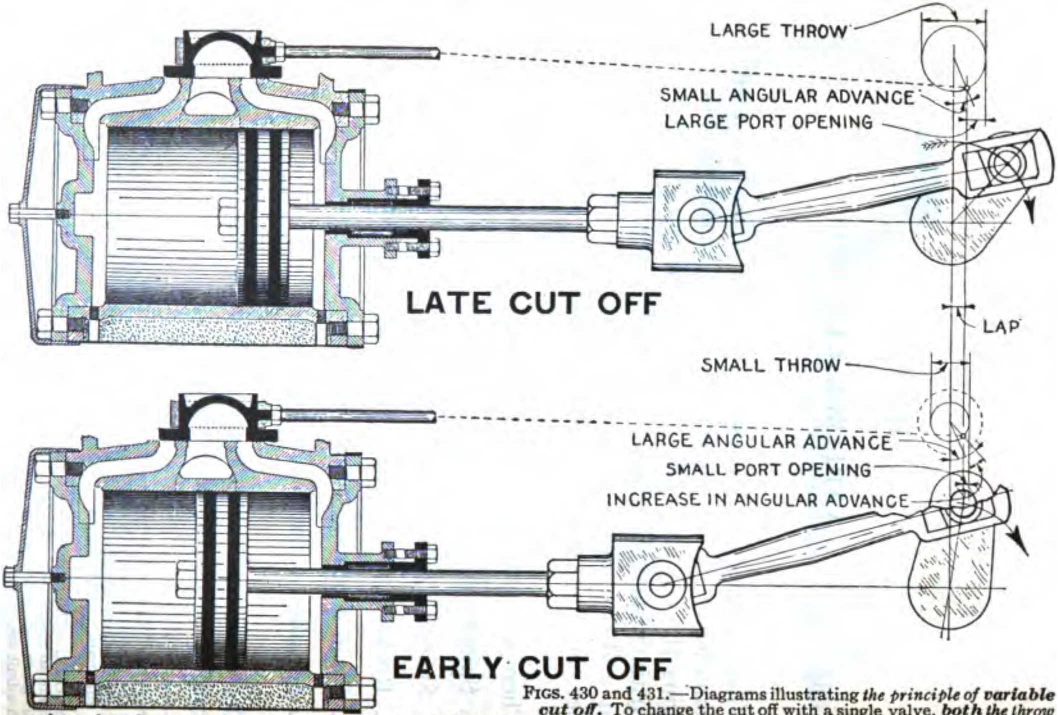
*NOTE.—The term automatic cut off engine is popularly applied to that large class of small and medium sized high speed engines having non-releasing valve gear; broadly speaking, any type of engine which adjusts itself to changes in load by automatically varying the cut off is an automatic cut off engine.

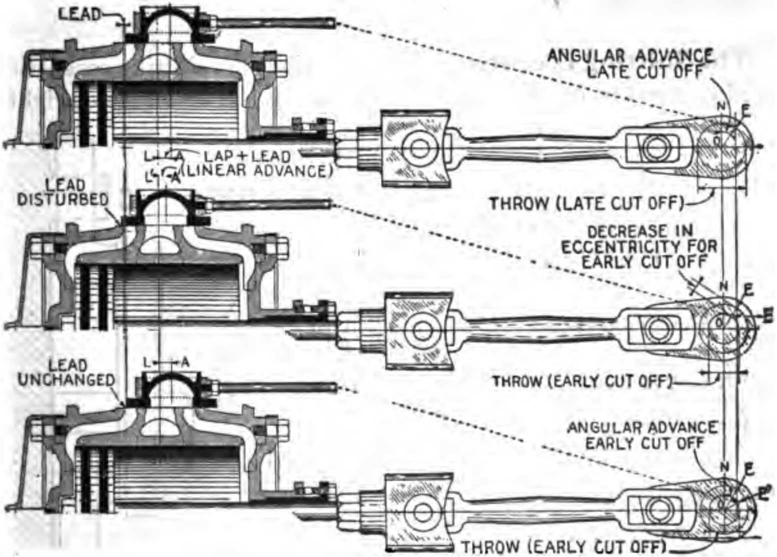
†NOTE.—The author objects to the term "expansion valve gears," because by usage it has come to mean *single variable expansion* gears as distinguished from *fixed expansion*, and *double* gears, all gears being expansion gears except a few, as for instance, pump gears, which admit steam for the full length of the stroke



FIGS. 430 and 431.—Diagrams illustrating *the principle of variable cut off.* To change the cut off with a single valve, ***both the throw and angular advance must be altered.*** This is the method of ***combined variable throw and variable angular advance*** and is the one employed on all single valve "automatic cut off" engines. It is frequently and erroneously called the method of variable travel, thus, the cut off is said to be shortened by reducing the travel. Figs. 432 and 433 show why this is wrong.

1. By single valve gear, or,
2. By double valve gear.

When a single valve gear method is employed, the cut off is called *variable*, as distinguished from the methods using double valve gear, in which case the cut off is said to be *independent*.



FIGS. 432 to 434.—Diagrams showing why *both the throw and angular advance* must be varied to change the cut off. In the figures let O, be the center of the shaft, and E, the center of the eccentric for maximum throw, then NOE, is the angular advance for maximum throw. Now, if *only* the throw be changed as in fig. 433, the center of the eccentric will be at some point E' on radius OE; evidently this reduces the linear advance LA, to L'A', thus disturbing the *lead*. Hence, when the travel is changed, as by reducing the *eccentricity* from OE to OE', figs. 432 and 433, the angular advance must be increased from NOE, to NOE'', fig. 434, in amount sufficient to maintain the linear advance constant in order not to alter the lead.

Principles of Variable Cut Off.—The cut off of the ordinary slide valve may be altered by *changing* **both** *the throw and angular advance*. In making these changes, *the* **shorter** *the travel*, *the* **earlier** *the cut off*. This way of changing the cut off may be called the method of *combined variable travel and variable angular advance*. There are two methods of moving the eccentric to vary both the travel and angular advance:

1. By shifting;
2. By swinging.

To distinguish the two constructions, the first is called the *shifting eccentric*, and the second the *swinging eccentric*, most engines being fitted with the latter.

The Shifting Eccentric.—The principle of a shifting eccentric is illustrated in fig. 435. A slot S, is cut in the eccentric at right angles to the crank; two arms A, B, project from the eccentric to the bearings, C, D, which are attached to the fly wheel, thus permitting the eccentric to "shift" at right angles to the crank.

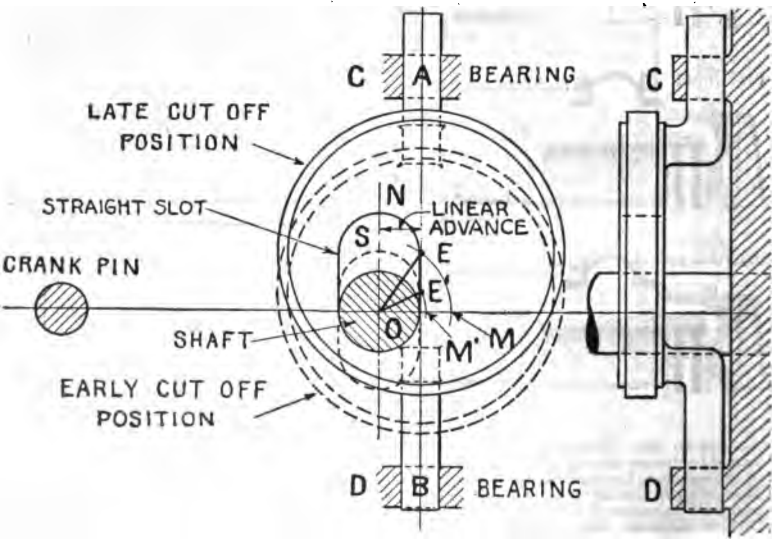
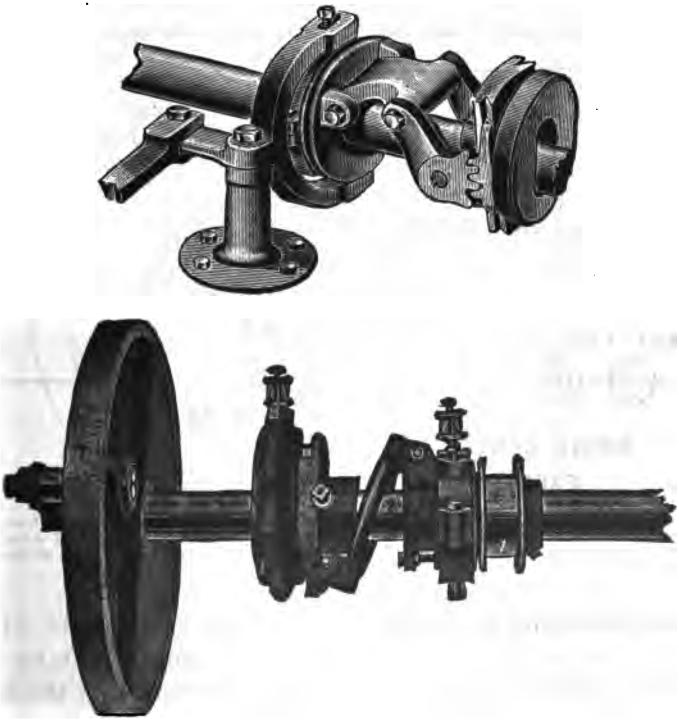


FIG. 435.—The shifting eccentric. Two arms A, B, attached to the eccentric, pass through the bearings C, D. The eccentric has a slot S to permit linear movement on the shaft. By *shifting* it from E to E', the throw is reduced, and the angular advance increased, the combined effect of which produces an earlier cut off. There is considerable movement and friction in the bearings C, D, as compared with the swinging eccentric. The lead is constant (not considering the angularity of the eccentric rod).

For a late cut off (full gear), the position of the eccentric is shown in full lines with its center at E. In this position O M, is equal to the eccentricity, or half the throw, and N O E, the angle of advance.

The cut off is shortened by ***reducing the throw and increasing the angular advance.*** This is done by ***shifting*** the eccentric along the slot to some intermediate position as that shown by the dotted lines with center at E′. This reduces the eccentricity, or half throw from O M, to O M′, hence, the travel of the valve has been reduced twice the distance M M′, and the cut off shortened an amount corresponding with the increase (E O E′) in the angular advance.



Figs. 436 and 437.—Some examples of shifting eccentrics as used on traction engines, for both variable cut off and reverse. Fig. 436, Heilman gear. As seen, the shifting eccentric slides in V, grooves, being geared to a double bell crank, which is connected by a link to a disc and collar arranged to slide along the shaft. A second bell crank and rod connects the collar with the control lever. Fig. 437 shows the Russell variable cut off and reverse. As constructed, the eccentric is made to shift for change of cut off or reverse by a single bell crank.

Since the center of the eccentric moves in a line E E′, at right angles to the crank, the lead remains constant.* The angular advance increases as the cut off is shortened. In moving the eccentric from E to E′, the angular advance increases from N O E to N O E′.

Ques. What is the action of a shifting eccentric in shortening the cut off?

Ans. It shortens the cut off *by reducing the throw and increasing the angular advance, sufficiently to maintain a constant linear advance.* (*not considering the eccentricity of the eccentric rod*).

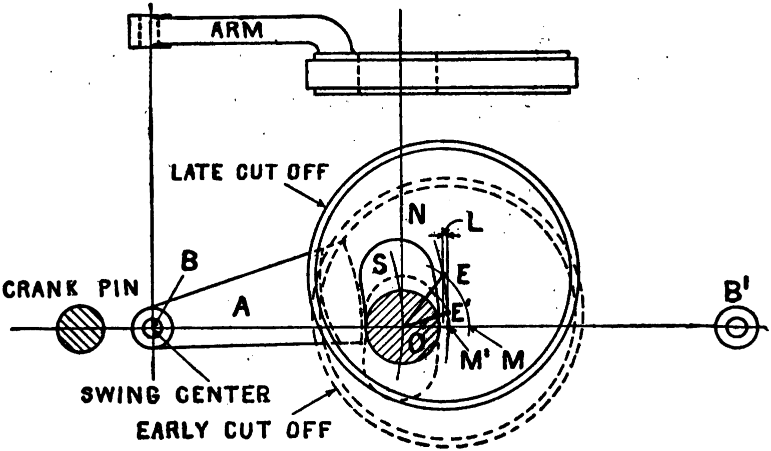


FIG. 438.—The swinging eccentric. The arm A, is pivoted at B, in line with the shaft and crank pin. This location of the swing center causes the lead to *increase* as the cut off is shortened. If the swing center be located on the opposite side at B′, the lead will decrease as the cut off is shortened.

The Swinging Eccentric.—The chief objection to the shifting eccentric is the friction brought on the bearings (C, D, fig. 435) which is liable to interfere with the free movement of the eccentric, and thus reduce the sensitiveness of the governor, especially on account of the difficulty of lubricating a bearing rotating around a center, accordingly the shifting eccentric is better adapted for an adjustable or non-automatic cut off engine.*

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*NOTE.—The effect of the angularity of the eccentric rod is to diminish the lead as the eccentric moves from E to E′, but since the rod is very long in proportion to the throw, the lead is only slightly reduced, hence for simplicity the angularity is neglected in the explanation.

To overcome this defect, the swinging eccentric has been devised by means of which the considerable linear motion through the bearings (C, D, fig. 435) has been reduced to a very small circular motion.

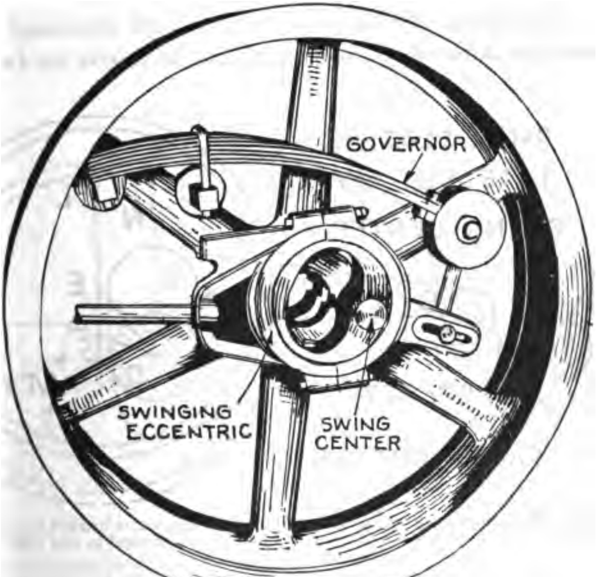


Fig. 439.—Fly wheel and governor of Buffalo engine illustrating the swinging eccentric. It should be noted that in the design here illustrated the swing center is near the radial slot instead of at the end of the arm as in fig. 438.

A swinging eccentric has one arm A, fig. 438, of any convenient length, and pivoted at some point as B, on a line joining the shaft and crank pin centers. The point B, is called the *swing center*. A circular slot S, is cut in the eccentric, having sides in the form of arcs of circles described with B, as center.

---

*NOTE.—Owing to the centrifugal force thus set up, the oil will not remain long in the bearing but is thrown off at the outer ends.

The action is similar to that of the shifting eccentric, that is, the cut off is shortened *by a reduction of throw and an increase of angular advance.* This is done by *swinging* the eccentric about the swing center B, from its full gear position E, to some intermediate position E′. Here, the angular advance has increased from N O E, to N O E′, and the throw reduced by twice the distance M′M.

Ques. What is the action of the swinging eccentric in shortening the cut off?

Ans. It shortens the cut off *by reducing the throw and increasing the angular advance in such proportion as to give an increasing lead.*

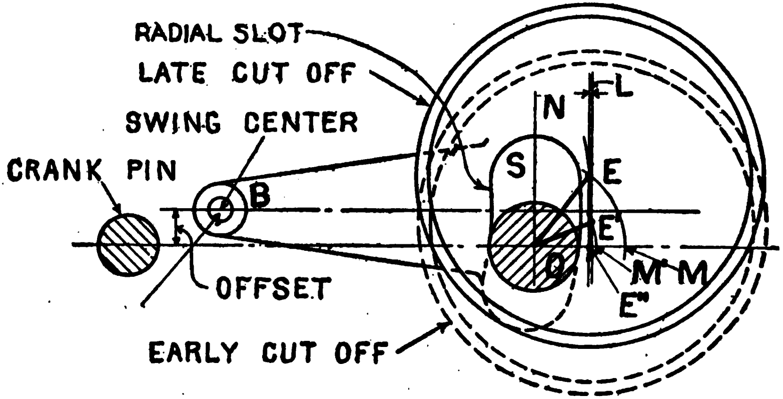


FIG. 440.—The offse *swinging eccentric.* With the swing center located as in the figure, the lead is the same for maximum and minimum cut off and greatest in mid position.

It should be noted in fig. 438, that as the center of the eccentric is moved from E to E′, to shorten the cut off, the lead is increased an amount equal to the distance L[*].

Ques. How would the action of the swinging eccentric be modified if the swing center be located on that side of the shaft opposite the crank pin?

Ans. The lead would *decrease* as the cut off is shortened.

---

[*] NOTE.—Taking into account the angularity of the eccentric rod, the actual increase in the lead is slightly less than the distance L.

This is objectionable in that it reduces the port opening as the cut off is shortened, thus producing wire drawing which lowers the admission pressure.

The Offset Swinging Eccentric.—On some engines the swinging eccentric is located with its swing center offset from the line joining the shaft and crank pin centers as shown in fig. 440.

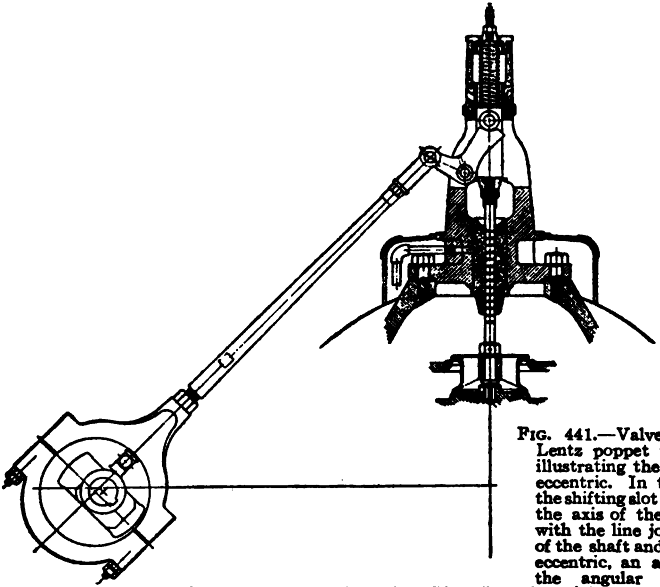


FIG. 441.—Valve gear of the Lentz poppet valve engine, illustrating the shifting type eccentric. In this eccentric, the shifting slot is cut straight, the axis of the slot making with the line joins the center of the shaft and center of the eccentric, an angle equal to the angular advance, the eccentric axis being in advance of the slot axis. The effect of straight slot eccentric when used for variable cut off is to give a constant lead for all degrees of expansion (not considering the angularity of the eccentric rod).

Ques. What is the object of offsetting the swing center?

Ans. To compromise between the conditions described in the last two examples, that is, instead of an increasing or decreasing lead, by offsetting the swing center *the same lead is obtained at both maximum and minimum cut off with a somewhat larger lead in mid position.*

In fig. 440, the swing center B, is offset above the crank axis one-half the distance from E, to this axis. This position B, gives the same lead in the two extreme positions, and it should be noted that the total increase of lead L, is only one-half the increase L, of fig. 438. The two positions illustrated, correspond to those of the two preceding figures showing the same angles of advance but less increase of lead. From the figure it is seen that if the eccentric be moved to the extreme position E″, the lead will decrease and become equal to the original amount for the full gear position E.

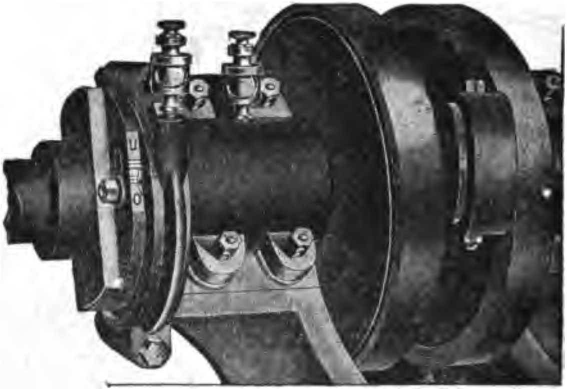


Fig. 442.—Leffel adjustable *shifting eccentric.* It consists of a hub plate keyed to shaft with valve eccentric bolted thereto, in a manner enabling adjustment of the cut off up to three-fourth stroke, or reversing motion of engine with same range of cut off.

Ques. What is the action of the offset swinging eccentric in shortening the cut off?

Ans. It shortens the cut off *by reducing the throw and increasing the angular advance, in such proportion that the lead* **increases** *for full gear to mid position and then* **decreases** *to the original amount at minimum cut off*, the total increase being less than that produced by the swinging eccentric.

Independent Cut Off.—For maximum economy, a much earlier cut off is required than that produced by a slide valve in full gear. For instance, a single cylinder engine to run with the least steam consumption per horse power must cut off from one-third to one-fifth when running non-condensing and from one-fifth to one-seventh when running condensing, the particular point depending upon the pressure and quality of the steam, etc.

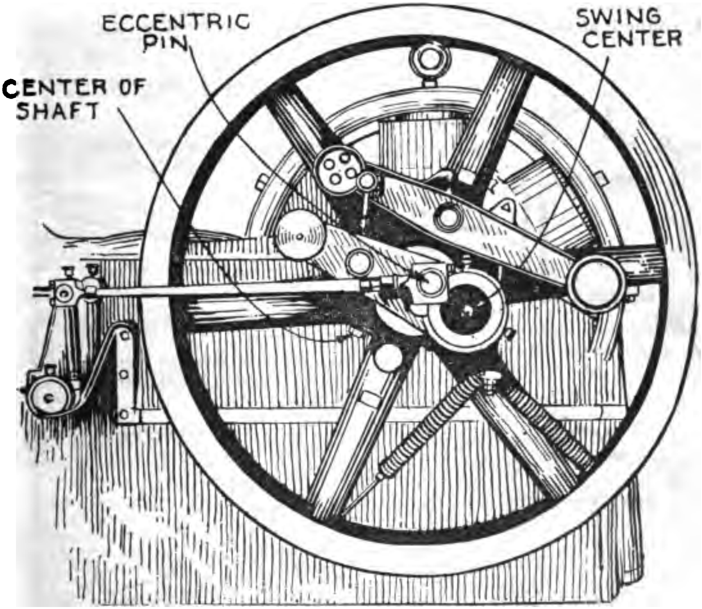
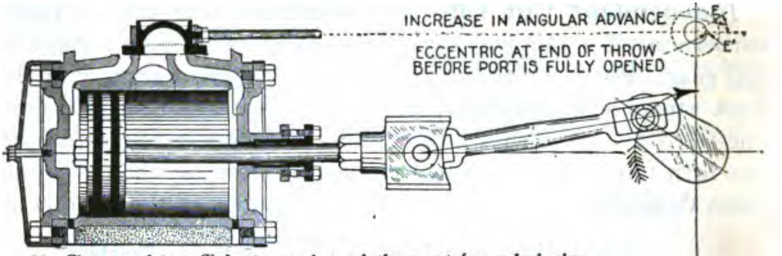


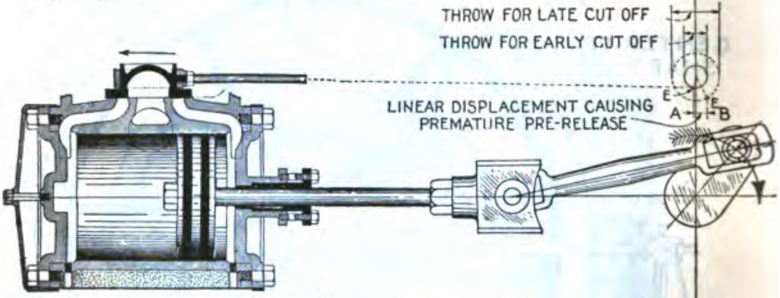
Fig. 443.—Wheel end of American Ball automatic cut off engine showing eccentric pin, largely used in place of an eccentric. The arms and spring are parts of the governor.

A considerable range of cut off is required on account of variations in power demands.

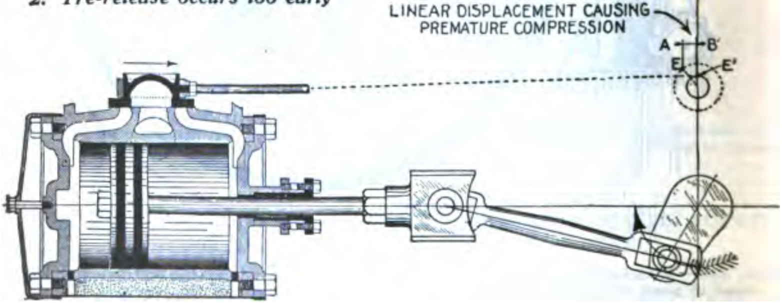
The plain slide valve is designed for the latest cut off required and with a movable eccentric the valve is made to cut off shorter


INCREASE IN ANGULAR ADVANCE
ECCENTRIC AT END OF THROW BEFORE PORT IS FULLY OPENED
E
E'

1. Slow and insufficient opening of the port for admission



2. Pre-release occurs too early

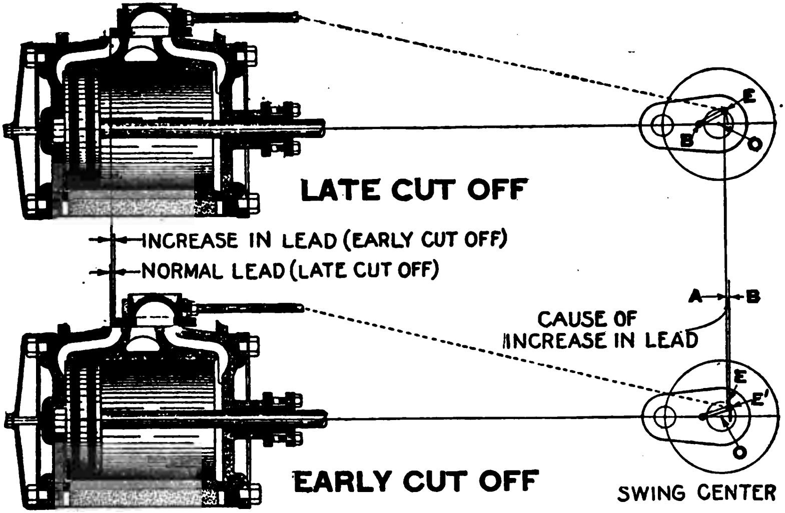


3. Compression begins too early

FIGS. 444 to 446.—*Defects of the slide valve at early cut off:* 1, fig. 444, *slow and insufficient port opening.* Note that the eccentric center E', is at the end of its throw; hence the valve movement in opening the port is comparatively slow; 2, fig. 445, *pre-release occurs too early.* This is due to the increased angular advance displacing the valve to the *left* by the distance AB; 3, fig. 446, *compression begins too early.* Similarly as in 2, the increased angular advance displaces the valve to *right* by the distance A'B', causing the valve to close to exhaust too soon. In the figures, E, is the center of the eccentric for full gear or late cut off, and E', for early cut off.

as previously explained. This combination has the advantage of simplicity but for very early cut offs it possesses certain defects which become more pronounced with the shortening of the cut off. These defects are, briefly:

1. *Slow and insufficient opening of the port for admission.*


E
B
O
LATE CUT OFF
INCREASE IN LEAD (EARLY CUT OFF)
NORMAL LEAD (LATE CUT OFF)
A
B
CAUSE OF INCREASE IN LEAD
E
E′
O
EARLY CUT OFF
SWING CENTER

FIGS. 447 and 448.—*Defects of the slide valve at early cut off:* 4, lead not constant with swinging eccentric. *Case I.* Swing center between shaft and crank pin. For early cut off, the center E, of the eccentric swings through the arc EE′, fig. 448, to position E′, thus increasing the angular advance and reducing the travel, but in so doing, the valve is displaced to the *right* a distance AB, *increasing* the lead by this amount.

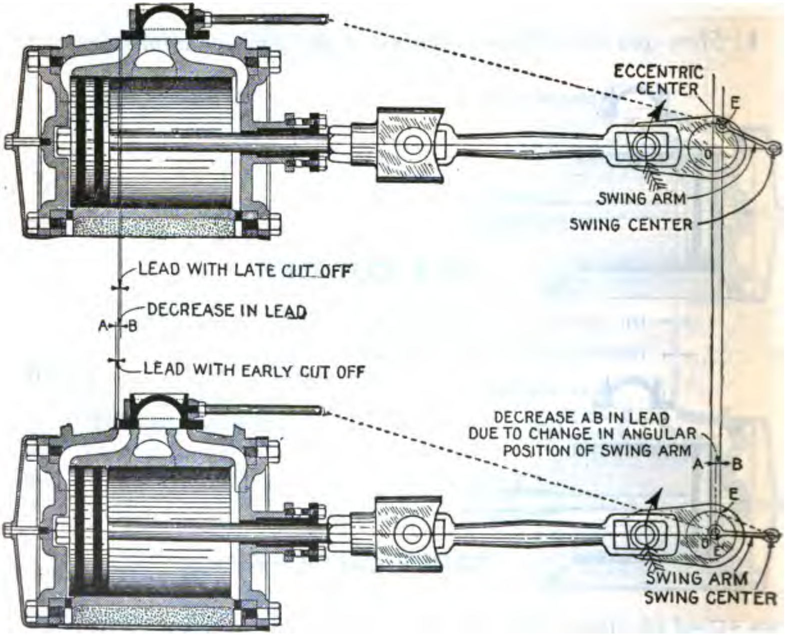
On account of the reduced travel, the valve moves slower at admission and cut off; this causes wire drawing at these points which together with insufficient port opening due to the small travel results in a loss of pressure.

2. *Pre-release occurs too soon.*

Since the valve opens to exhaust sooner than is necessary, the full benefit which might be derived from expansion of the steam is not realized.

3. *Compression begins too early.*

This produces a resistance in excess of that required to overcome the momentum of the reciprocating parts. The slower the engine speed, the more pronounced is this effect.



FIGS. 449 and 450.—***Defects of the slide valve at early cut off; 5, lead not constant with swinging eccentric. Case II,*** **swing center and crank pin on opposite sides of shaft. When the center of the eccentric swings through the arc EE', fig. 450, to shorten the cut off, the valve is displaced to the *left* a distance AB, thus decreasing the lead by this amount.**

4. *The lead is not constant* (with swinging eccentric). The variation of lead is influenced chiefly by the position of the swing center, and also by the length of the swing radius.

The independent cut off is intended to overcome these defects and consists of:

1. A *main valve* which controls the points of admission release and compression, and
2. A *cut off valve* which controls the cut off.

There are two eccentrics, one for each valve. The main valve is operated by a fixed eccentric, and the cut off valve by a *rotating eccentric*. With this combination, the cut off may be varied without changing the positions of release and compression.

Ques. Where is the cut off valve located?

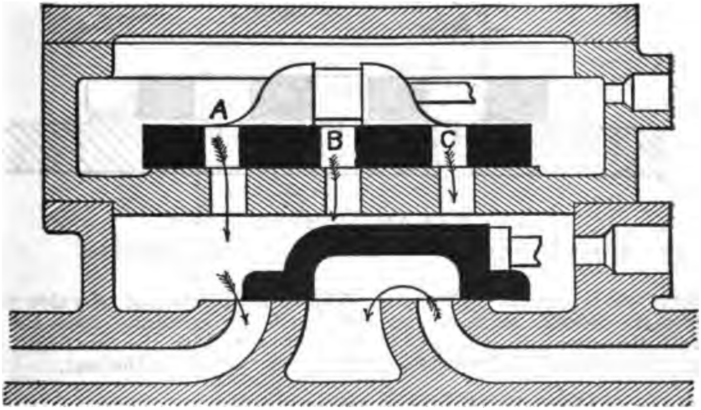


FIG. 451.—The Gonzenbach independent cut off valve. This is located in a separate steam chest above the main valve, the latter being an ordinary slide valve which controls the steam distribution with the exception of cut off. The range of cut off is limited, and the lower steam chest presents a large clearance which is objectionable. Moreover, the main valve is inaccessible.

Ans. It may be, 1, placed in a separate steam chest, or 2, arranged to work on the back of the main valve.

Ques. What is it called when arranged to work on the back of the main valve?

Ans. A riding cut off.

Ques. Describe the type with separate steam chest.

Ans. The Gonzenbach valve shown in fig. 451 is an example of this type. In the figure, the main valve, which is the lower, is an ordinary slide valve; the cut off valve which works on a ported partition directly above, is of the gridiron type, that is, there are a number of steam ports (A, B, C,) in order to secure a quick cut off with moderate travel.

During admission, steam passes through the ports A, B, C, into the lower steam chest and to the cylinder through either one of the cylinder ports which happens to be open. The action of the cut off valve differs from the ordinary valve in that while the latter opens and closes the port with the same edge, the cut off valve does this with the two edges, that is, the port in the valve passes bodily across the port in the seat.

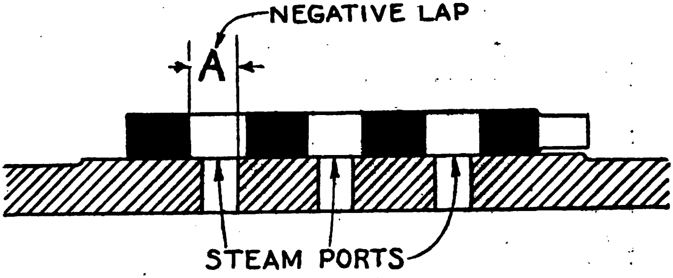


FIG. 452.—Gonzenbach cut off valve in neutral position showing negative lap.

Ques. How is the cut off varied?

NOTE.—If the travel of the valves on a locomotive for full gear be 4½ to 5 inches, for a lead of 1/16 inch at full gear, and 5/16 inch at mid-gear, a steam lap of ¾ inch and no exhaust lap will secure excellent results; but if the engine were never to run at a speed of over 20 miles an hour, an exhaust lap of ¼ inch could be used to advantage. Most builders of stationary engines give so much exhaust lap that a considerable back pressure is caused, and the engines can not be run at high speed, and for two reasons: 1st, that the steam does not get out of the cylinder fast enough, and, 2nd, there is not enough cushion to take up the momentum of the connections at high speed. An early release and strong cushion are required for high speeds. At moderate speed an early release and strong cushion deaden the motion of the engine over the centers, and the use of two slide valves, one on top of the other, was suggested by Meyer. A false valve seat was suggested by Rankine, with the object of obtaining a quicker cut off, the seat being moved by one eccentric while the valve was moved by another. In this way the effect of an eccentric with greater throw was obtained. The first change suggested by Gonzenbach consisted in making the steam chest in two chambers. In the one next the cylinder, the ordinary slide was employed while the steam came in through openings from the other chamber, these openings were covered by a simple slide moved by an eccentric. Thus the inlet and exhaust were regulated by the ordinary slide, but the second one cut off the supply of steam. As the principal objection to this was the large clearance space left in the main steam chest and the consequent waste of steam, the Meyer gear became the favorite.

Ans. By turning the cut off valve eccentric forward or backward on the shaft as the case may be.*

Fig. 452 shows the cut off valve in its neutral position from which it is seen that the valve has negative lap. This may equal or exceed the width of the ports in the seat; the negative lap being the distance A, measured from one edge of the seat port to the opposite edge of the valve port.

The principles of the Gonzenbach valve are best understood by the application of the Bilgram diagram.

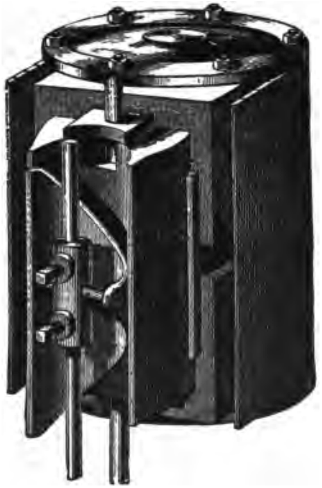


Fig. 453.—Rider variable cut off gear. In this riding cut off gear, the cut off is altered, as in the Meyer gear by varying the lap. *In construction,* the back of the main valve is hollowed into a part of a cylinder whose axis is the center of the riding valve. The lap edges of the riding valve and steam edges of the main valve are tapered in such a manner that by rotating the riding valve, its lap is changed. This rotation is accomplished by a spindle attached to the governor. gearing into a sector on the valve stem.

Problem.—In a Gonzenbach valve gear the main valve has $\frac{1}{16}$ lead; $\frac{1}{2}$ inch port opening, and cuts off at $^6/_{10}$ stroke; the cut off valve has 3 ports giving $\frac{1}{2}$ inch port opening each. Required the negative lap and positions of the cut off valve eccentric for the earliest and latest cut off.

***NOTE.—The effect of this is to change the angular advance which alters the cut off. The expansion eccentric, unlike the shifting or swinging eccentric, does not reduce the travel in changing the cut off.**

In fig. 454 crank position C, for latest cut off, the center E_M, of the main eccentric, and lap of the main valve are found in the usual way.

The lead position is now found by drawing the line LL′, through O, and tangent to the main valve lap circle.

At the latest cut off ($^6/_{10}$ stroke), the negative lap circle must be tangent to the line OC, and also tangent to the lead position OL.

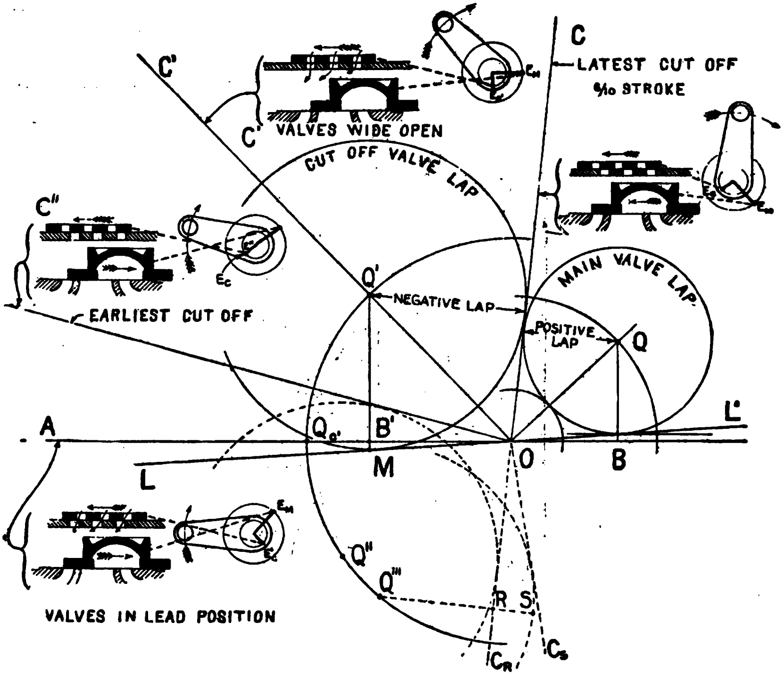


FIG. 454.—Application of the Bilgram diagram to the Gonzenbach independent cut off gear. The relative positions of the valves and eccentrics are shown for four crank positions which illustrates the operation of the gear.

Since the cut off valve has three ports, it is equivalent to a single valve having three times the port opening and travel. Hence, with radius of 1½ inch equal to three times the port opening of each port of the cut off valve, describe the negative lap circle tangent to OC and OL, which gives the position Q′, of the cut off valve eccentric for latest cut off.

Starting at crank position A, and with the cut off eccentric at Q′, both valves are open to lead; the main valve is at a distance QB, from its neutral position, and the cut off valve at a distance Q′B′. This latter distance being less than the negative lap, the cut off ports are open an amount B′M equal to the lead.

As the crank moves, the cut off ports are further opened, up to position C′, where they stand wide open as shown, this being the neutral position for the cut off valve.

As the crank advances further, the cut off begins to close the port, ***not by changing its direction*** as in the case of the slide valve but by ***continuing its movement*** to the end of its travel. Cut off occurs at crank position C.

The earliest point of cut off is determined by extending CO, downward and finding Q″, such as to make the negative lap circle tangent to CO, extended, and drawing OC″, tangent to the negative lap circle which gives the crank position for earliest cut off. After passing this point expansion takes place both in the cylinder and the lower steam chest until the main valve cuts off at position OC, where it is continued in the cylinder alone until pre-release.

Ques. Mention the defects of the Gonzenbach valve.

Ans. In shortening the cut off with this valve gear admission to the lower steam chest occurs earlier and earlier, hence there is a point beyond which the cut off valve would admit steam to the lower steam chest before the main valve had closed its port to admission in the previous stroke, thus admitting steam to the cylinder twice during the stroke.

Thus in fig. 454, if the cut off eccentric be advanced to Q‴, the negative lap circle will cut OC, extended, indicating that the cut off valve ports were open a distance RS, when the main valve closed on the previous stroke, thus readmitting steam to the cylinder from crank positions C_s, to C_r, during the expansion period of the previous stroke.

This limits the range of cut off, and in order to secure an earlier cut off, it is necessary to design the main valve for shorter cut off. The range of cut off is therefore limited. Moreover, on account of the large clearance of the lower steam chest, the full expansion due to the cut off is not secured, the difference between the apparent and real expansion increasing for early cut offs. An additional defect is that the main valve is inaccessible.

The Gonzenbach valve, on account of these objections is not extensively used, however, it serves to make clear the general principles of independent cut off.

The Riding Cut Off.—The large clearance and inaccessible main valve of the Gonzenbach gear are overcome in the riding cut off by placing both valves in one steam chest and using the back of the main valve as a seat for the cut off valve, that is, the cut off valve "rides" on the main valve, hence the name *riding cut off*.

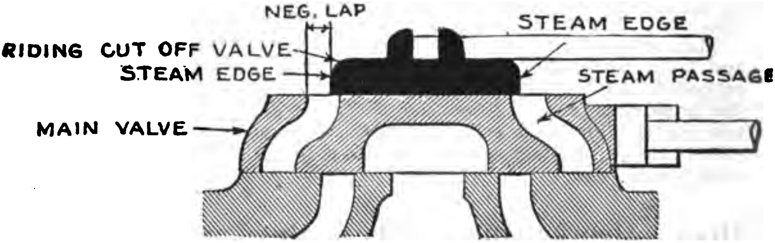


FIG. 455.—Riding cut off valve with outside cut off edges. ***In operation*** the cut off valve travels or "rides" on top of main valve, and with fixed lap as above, receives its movement 1, from a rotating eccentric, that is, an eccentric loosely journalled on the shaft so that its angular advance may be changed to vary the cut off, under control of: 1, a governor, or 2, a fixed eccentric with link motion. ***The riding cut off valve with outside cut off edges gives quickest cut off with early cut offs.***

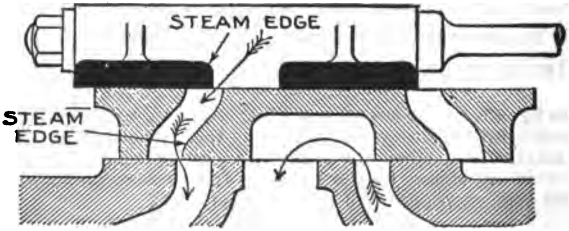


FIG. 456.—Riding cut off valve with inside cut off edges. ***This arrangement having inside cut off edges gives quickest cut off with late cut offs.***

Fig. 455 shows a simple form of riding cut off. Both valves are shown in neutral position, in order to show the ***positive*** lap of the main valve, and ***negative*** lap of the cut off valve.

The main valve is nothing more than the ordinary slide valve having steam passages in the end leading to the back which is machined to form the seat for the cut off valve.

Fig. 456 shows a riding cut off valve which cuts off at the inside edges of the ports; it necessarily has considerable positive lap.

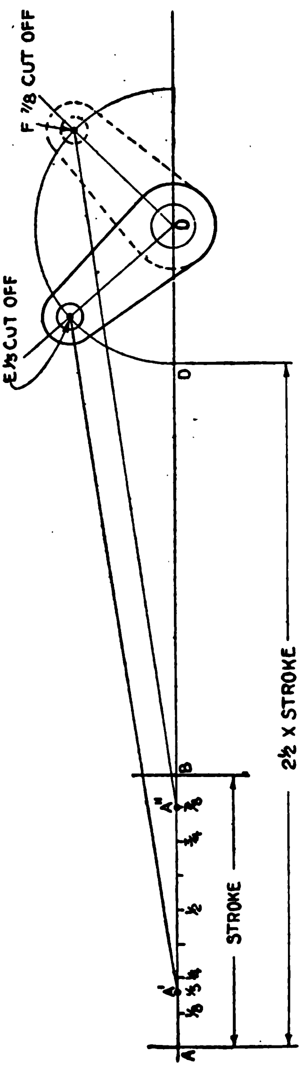


FIG. 457.—Angular position of crank for 1/5 and 7/8 cut off. With any convenient scale lay off A B, on a horizontal line = stroke, and divide it into fractional parts. Locate O, so that A O = 3 × A B. With O, as center and O D, as radius = ½ stroke describe the semi-circle; A D, then, will be the length of the connecting rod for the given ratio 2½ : 1. With A′, at 1/5 stroke as center, and radius = A D, describe an arc cutting the semi-circle at E, giving crank position for 1/5 cut off. Similarly with same radius and center at A″, (7/8 stroke) describe arc cutting at F, giving crank position for 7/8 cut off.

Methods of Variable Cut Off with Riding Valve.—Cut off by a riding valve may be varied in three ways, as by:

1. Variable angular advance;
2. Variable lap;
3. Variable travel.

The first method employs a rotating, or loosely journaled eccentric for the cut off valve whose angular advance is controlled by a governor.

The second method has a fixed eccentric to operate the cut off or "riding" valve, the lap of the latter being adjustable by a right and left screw.

The third method employs a link to vary the travel of the riding valve.

For convenience, the eccentric which operates the riding valve is called the *riding eccentric**; it is called by some writers the cut off eccentric, and more commonly, though ill advisedly, the *expansion eccentric.*

1. Riding Cut Off; Variable Angular Advance.—The usual range of angular advance given to the riding eccentric is from a little less than 90° up to 180°. The relative positions of the two eccentrics are such that the valves move in opposite directions at cut off (within limits) for a well designed gear.

---

*NOTE.—The term *riding eccentric* is used for the loosely journaled eccentric connected to the riding valve to distinguish it from the fixed or *main eccentric* which operates the main valve. The symbols E_r and E_m being used to respectively designate each.

The following example will serve to illustrate the features of this gear.

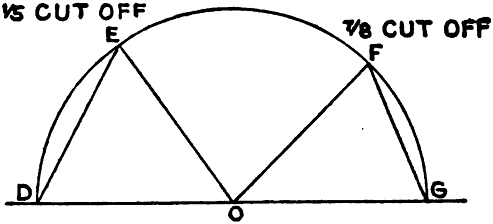


Fig. 458.—Method of transferring crank positions determined for given cut off. In order not to complicate the Bilgram diagram all unnecessary lines should be omitted, hence the crank positions for given cut offs are best determined in a separate diagram as in fig. 457 and then transferred, as above. In the figure draw a horizontal line, and with radius = O D, of fig. 457, describe the semi-circle D E F G. With D, as center, and radius = distance D to E, of fig. 457 describe an arc cutting the semi-circle at E, giving crank position O E, for ⅕ cut off. A similar construction gives O F, crank position for ⅞ cut off. After locating O E and O F, the semi-circle and radii D E and G F, should be erased leaving only the horizontal line and the two cut off positions O E and O F, to be used in the Bilgram diagram fig. 459.

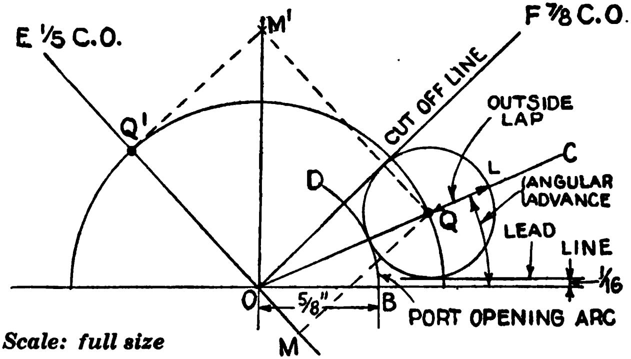
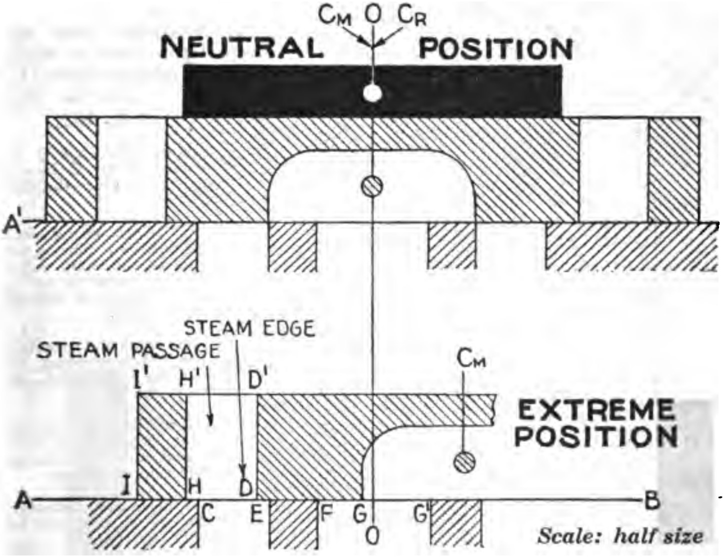


Fig. 459.—Bilgram diagram for main valve. Here the lap, angular advance, and travel are determined for ⅕ cut off and ⅝ port opening as explained in the accompanying text.

Example.—Determine the dimensions of a riding cut off that will meet the following requirements: maximum cut off 7/8 stroke; lead 1/16 in.; ports, 3/4 in.; port opening not less than 1/2 in. for any cut off up to 1/8 stroke. Bridges 1/2 in.; connecting rod ratio 2½ : 1.

1. Find crank position for 1/8 and 7/8 stroke as in fig. 457;

2. Transfer the crank positions for 1/8 and 7/8 cut off just found to fig. 459, *as explained in fig.* 458;



FIGS. 460 and 461.—Detail of main valve and seat. On A B, lay off the steam port C E = 3/4 in., the port opening C D = 5/8 in., and the bridge E F = 1/2 in. Sketch in valve end in extreme position. D, will be the steam edge. Lay off D H = 3/4 in. and draw D D′, and H H′, giving the steam passage through the valve. This is usually made same size as the port to reduce friction. I I′, the end of the valve is located far enough beyond H H′, to give a steam tight joint, say 1/2 in. Locate G, the exhaust edge of the valve, so that D G = *lap* + *port* = 5/8 in. + 3/4 in. The edge G′, of the bridge is so located that G G′ = C E. The center line O O, is now drawn half way between F and G′. Transfer the detail of valve end thus found to A′ B′, showing it in neutral position, and complete the valve and seat as shown.

3. Find outside lap and travel of main valve for 1/16 lead, 7/8 cut off, and a trial port opening of say, 5/8 in;

In fig. 459 draw the lead line 1/16 in. above, and parallel to the horizontal line. With radius O B = 5/8 in. port opening, describe the port opening arc B D.

NOTE.—In this chapter, O O, = center line of valve seat; C_m = center line of main valve; C_r = center line of riding valve; E_m = center of main eccentric; E_r = center of riding eccentric; E_v = center of virtual eccentric.

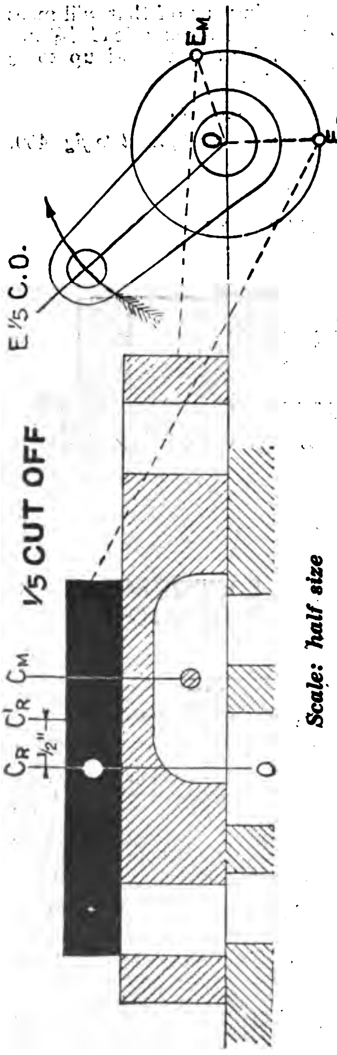


FIG. 462.—Position of main and riding valves corresponding to crank position for 1/5 cut off. At the right is shown the crank position O E, and center of main eccentric E_m, whose angular advance is obtained from fig. 459. E_r, is the center of the eccentric of the riding valve. Both valves are shown in the positions corresponding to those of the eccentrics, C_r, being the center of the riding valve, and C_m, the center of the cut off valve. Clearly, the valves are displaced with respect to each other the distance $C_r C_m$. The main valve is moving to the right, and the riding valve to the left. From the position of the eccentrics it is clear that the main valve is practically at rest and the riding valve traveling at maximum speed.

Describe, with a center Q, and radius found by trial the *lap circle* tangent to cut off line O F, lead line and port opening arc B D. Q L, then = outside lap, and O Q = half the throw, or eccentricity.

4. ***Draw main valve and seat as in figs.* 460 *and* 461;**

5. ***Determine from fig.* 459 *displacement of main valve for crank position* O E, *and draw valve in this position as in fig.* 462;**

In fig. 459 draw Q M perpendicular to O E extended which gives the displacement.

Fig. 462 shows the valve in this position. This is the position of the main valve when the crank is at 1/5 stroke, the point of cut off by riding valve.

The crank and eccentric position corresponding to that of the main valve are shown at the right.

It will be observed that the valve is practically fully open and about to change its direction of travel.

If the angular advance of the riding eccentric be made Q', fig. 459, for 1/5 cut off, such that the riding valve will be in the neutral position, that is Q', will be on O E, the riding valve will be traveling at its maximum velocity, tending to give a sharp cut off which is to be desired.

In fig. 459, the main valve for crank position O E (1/5 cut off) is displaced the distance Q M, or its equal Q' M'. Now, if the riding valve had zero lap, the main valve port would still be covered by the distance Q' M'. Hence, if cut off is to occur at O E (1/5 stroke), the riding valve must have a *negative* lap equal to Q' M'.

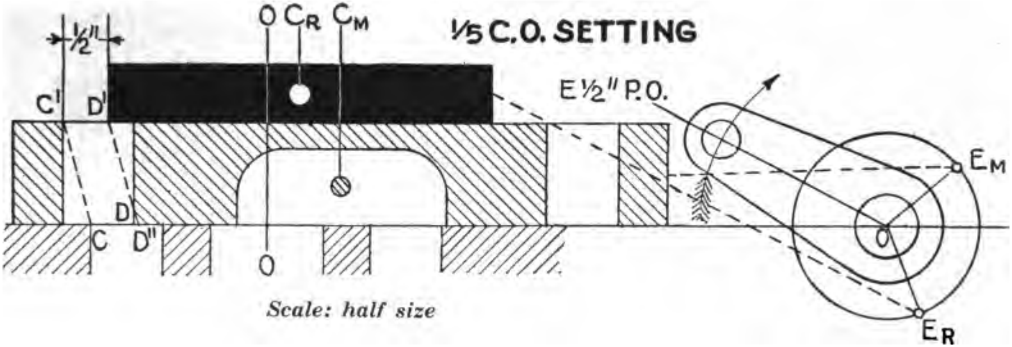


FIG. 463.—Position of valves when the riding valve has opened the port a distance C′ D′ = ½ in. with ¹/₅ cut off setting. Draw the dotted parallel lines C′ C and D′ D″, then C′ D′ = C D″, from which it is seen that the main valve has opened a distance C D, *less* than C D″. Hence, the ***effective*** port opening is less than ½ in. for the cut off setting of the riding eccentric, and the design accordingly does not meet the requirement. ***Some modification*** must be made, as explained in the accompanying text.

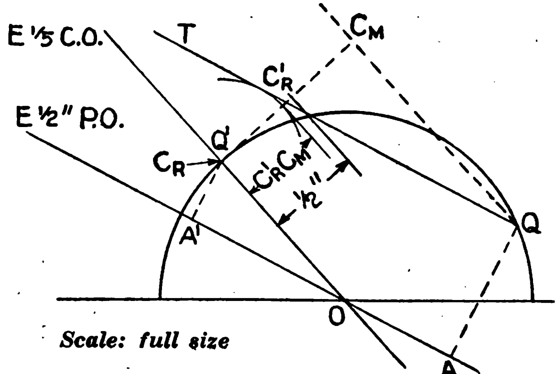


FIG. 464.—Diagram for obtaining crank position corresponding to ½ in. port opening of riding valve for ¹/₅ cut off setting. Transfer from fig. 459, Q, Q′ and crank position O E, for ¹/₅ cut off. Through Q, draw a line parallel to O E, ¹/₅ C. O, and at Q′, erect a perpendicular giving Q′, C_m, which represents the displacement of the main valve for crank position O E, ¹/₅ C. O. the riding valve being in the neutral position. Evidently Q′, corresponds to C_r, and C_m, to C_m, in fig. 462. For maximum port opening of ½ in. of the riding valve, $C'_r C_m$ (in fig. 462) = $C_r C_m$ — ½, that is, the crank must be turned backward from the ¹/₅ cut off position (fig. 462) until the valves are ½ in. nearer together. Hence, with C_r, as center and radius = $C_r C_m$ — ½ in., or $C'_r C_m$, describe an arc and draw a tangent Q T. Through O, draw O E ½″ P O, parallel to Q T, then O E ½ P O, is the crank position for ½ port opening of the riding valve.

6. *Determine if the port opening for 1/5 cut off be not less than the limit 1/2 in.;*

Fig. 462 shows the positions of the two valves at 1/5 cut off, their centers being displaced a distance $C_r\,C_m$. If the crank be rotated backward, the port will be opened by a movement of the main valve to the left, and a movement of the cut off valve to the right. Hence, for position of 1/2 port opening the crank must be rotated backward until C_r, reaches the position C'_r, with respect to C_m, such that $C'_r\,C_m = C_r\,C_m - \frac{1}{2}$ in. This position is obtained as in fig. 464, the valves and positions of eccentrics being shown in fig. 463. The figure shows that the port opening C D, of the main valve is less than C′ D′, of the riding valve by an amount equal to D D″. At no point of admission does the port opening become 1/2″ because, as is evident, if the crank be turned backward the main valve closes; if forward, it opens, but the riding valve closes, hence, the design must be modified to obtain not less than 1/2″ port opening through both valves.

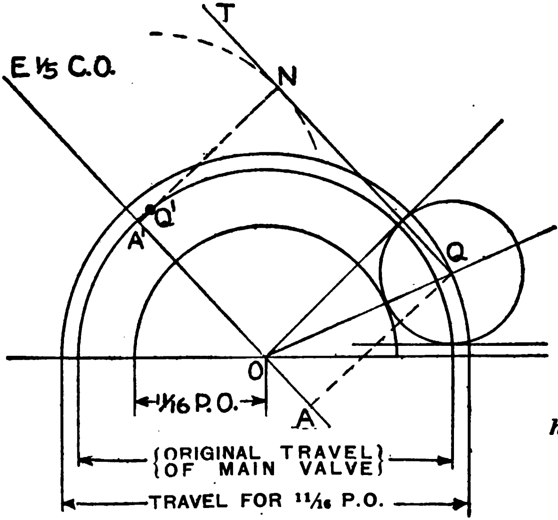


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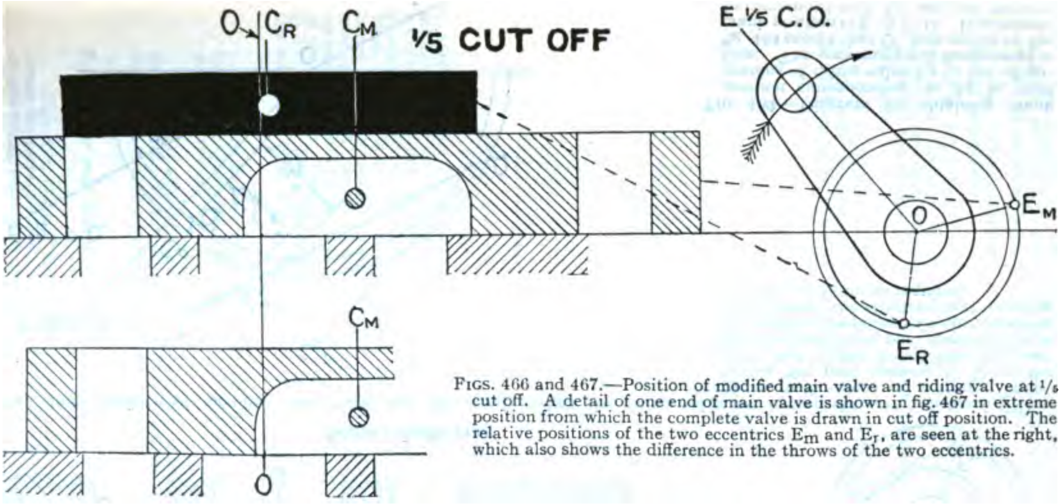
FIG. 465.—Bilgram diagram for modification of main valve and riding eccentric setting to give 1/2 in. *effective* port opening for 1/5 cut off. The diagram is constructed in the usual way giving Q, for the modified main eccentric, and the new lap circle. Through Q, draw Q T, parallel with 1/5 cut off crank position O E 1/5 C. O. If the original negative lap of the riding valve be retained, its travel will remain the same but the angular advance will be slightly changed. With *radius = the negative lap* and center on the smaller travel semi-circle, describe an arc tangent to Q T, giving Q′, the new position of the riding eccentric. For position O E 1/5 C. O., the main valve is displaced a distance Q A, hence the riding valve to cut off at this point must be displaced this distance minus its negative lap that is Q A — Q′ N, or Q′ A′.

7. *Modify design for 1/2 in. port opening at 1/5 cut off;*

This may be done by re-designing the main valve for a larger port opening, say 11/16, as in figure 465.



FIGS. 466 and 467.—Position of modified main valve and riding valve at 1/5 cut off. A detail of one end of main valve is shown in fig. 467 in extreme position from which the complete valve is drawn in cut off position. The relative positions of the two eccentrics E_m and E_r, are seen at the right, which also shows the difference in the throws of the two eccentrics.

The diagram, fig. 465, gives the proportions for the increased port opening, and fig. 466 and 467, the layout of valve showing both valves in position for 1/5 cut off.

The diagram, fig. 469, shows crank position E ½ P. O., for ½ in. port opening of the riding valve when the riding eccentric is set for 1/5 cut off, and fig. 468, the valve in this position. Here it is seen that the port opening C D, of the main valve is as large as C′ D′, of the riding valve, thus giving an effective port opening of ½ in. for the 1/5 cut off setting, the condition required.

It will be noticed in fig. 465, that increasing the port opening of the main valve involved increasing its travel and decreasing the angular advance of the riding eccentric.

8. *Show modified design in position of latest and ½ cut off ~~and note its~~ characteristics;*

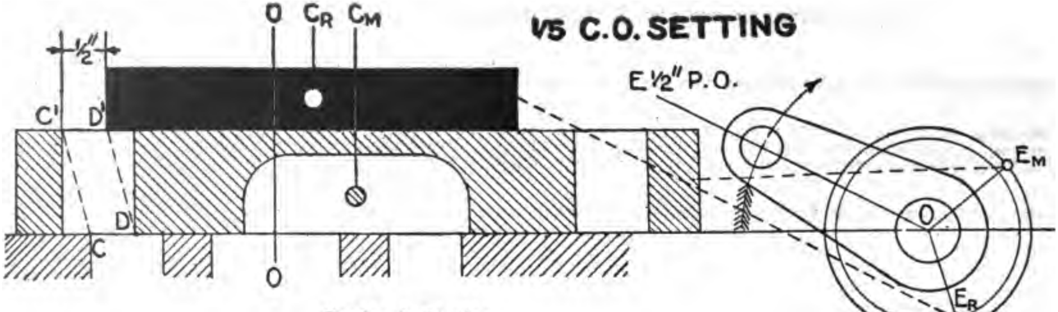


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FIG. 468.—Position of modified main valve and riding valve when the riding valve has opened the port a distance C′D′= ½ in. with 1/5 cut off setting. Transferring the port opening C′ D′, to the valve seat by the dotted lines it is seen that the main valve port opening C D, is the same as the riding valve port opening C′D′, hence, the effective port opening is C′ D′, or ½ in. as required.

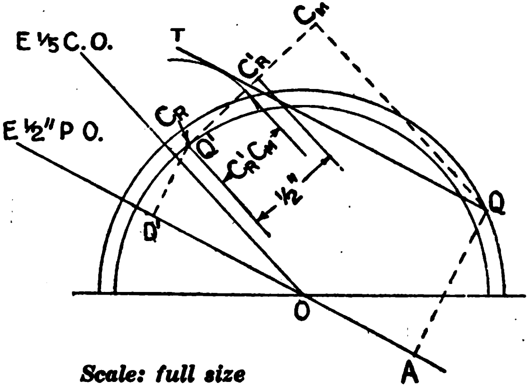

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FIG. 469.—Diagram for obtaining crank position corresponding to ½ in. port opening of riding valve for 1/5 cut off setting. The construction is the same as in fig. 464 except that Q′, does not lie on the crank position O E 1/5 C, O. It is lettered to correspond with fig. 464 and requires no further explanation.

The diagram, fig. 470, gives the displacement of the valves for the two cut offs, and figs. 471 and 472, the crank positions for ½ port opening of the riding valve for the ⅞ and ½ cut off settings of the riding eccentric.

Figs. 473 and 474 show the positions of the valves corresponding to the ⅞ and ½ cut off respectively.

Fig. 473 indicates that the later cut offs are "sluggish," the valves traveling in the same direction at almost the same speed. From the positions of the eccentrics shown at the right, evidently, the main valve is traveling the faster, hence the riding valve will reopen the port, but as the main valve cuts off at this point, re-admission will not occur.

Fig. 474 shows a sharper cut off at ½ stroke. Here both valves are still traveling in the same direction, the main valve being almost stationary while the riding valve is traveling at about maximum velocity.

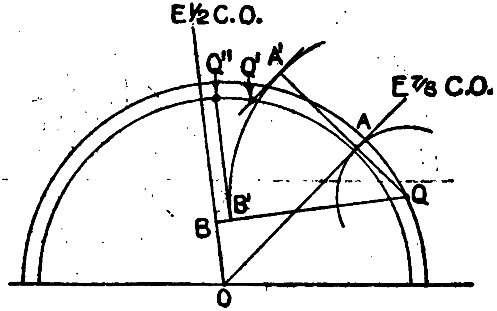

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Fig. 470.—Diagram for ⅞ and ½ cut off of the riding valve. Evidently for any cut off, the displacement of the riding valve is equal to *the displacement of the main valve* **less** *the negative lap of the riding valve*. Hence, describe an arc about Q, equal to the negative lap Q A′, and draw a tangent parallel to the given cut off position of the crank, cutting the riding travel circle. Thus, for O E ⅞ C. O., the main valve is displaced a distance Q A. If the riding valve is to cut off at this point it must be displaced **from** *the main valve an amount equal to its negative lap*, or Q A′, that is, it is displaced a distance A A′, on the other side of the neutral axis corresponding to O C_T, in fig. 473. The tangent through A′, gives Q′, the corresponding position of the eccentric. Similarly for ½ cut off, Q B, is the displacement of the main valve, and Q B — Q B′, or B B′, displacement of riding valve, Q″ being the corresponding center of the eccentric. Here B B′, corresponds to O C_T, and B Q, to O C_m, in fig. 474.

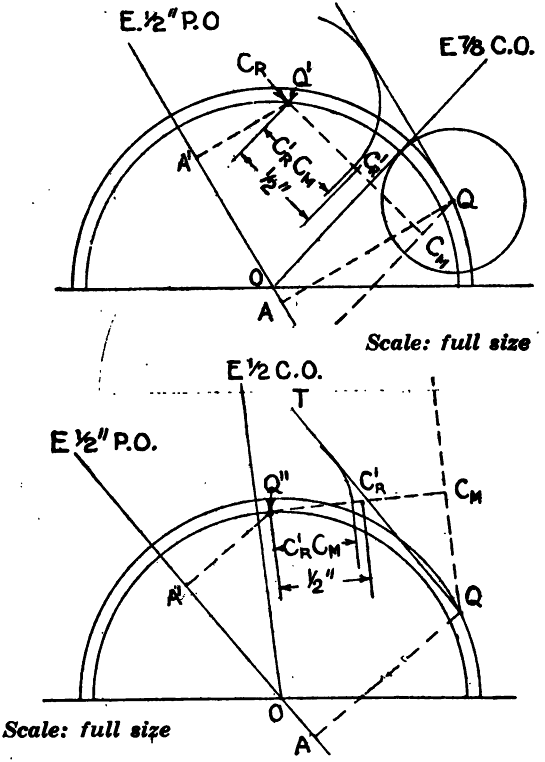
Fig. 466 shows a still sharper cut off at $^1/_8$ stroke, the valves in this case moving in *opposite* directions, the main valve being almost at rest, and the riding valve, at maximum velocity.

Figs. 475 and 476 show position of valves for ½ in. port opening of the riding valve for the ⅞ and ½ cut off settings of the riding eccentric. The port opening of the main valve in each case being greater than ½ in. but less for ½ than for ⅞ cut off.

9. *Test for over travel of the riding valve*;

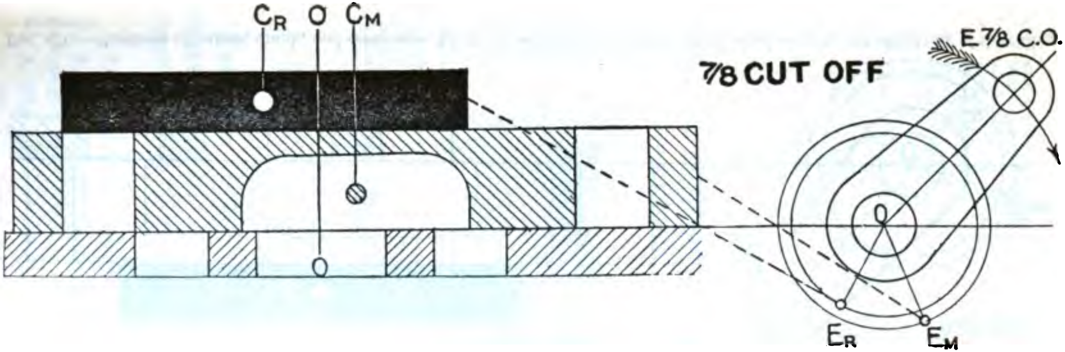
Figs. 473, 474 and 469 show that the earlier the cut off, the greater the angle between the two eccentrics. In other words the shorter the cut off, the greater the travel of the riding valve with respect to the main valve regarding the latter as stationary.

Hence, assuming no overtravel of the riding valve with respect to the main valve, at latest cut off, it is desirable to know at what cut off overtravel begins, because the constant running of the engine with undertravel would cause the riding valve to wear a "shoulder" on the main valve, causing leakage and perhaps a knock at short cut off. Unless there be overtravel for cut offs near the working cut off ($^1/_5$ in this case), the design should be further modified and the range of overtravel made as great as possible.



FIGS. 471 and 472.—Diagrams for obtaining crank positions corresponding to ½ in. port opening of the riding valve for ⅞ and ½ cut off settings of the riding eccentric respectively.

Regarding the main valve as stationary, the travel of the riding valve on the main valve may be regarded as obtained from an **imaginary eccentric** *of such throw and angular advance as to duplicate the movement of the riding valve with respect to the main valve as obtained from the two eccentrics;* such imaginary eccentric or radius is called **the virtual eccentric**, from which the cut off setting at which overtravel begins is



FIGS. 473.—Positions of valves and eccentrics for ⅞ cut off.

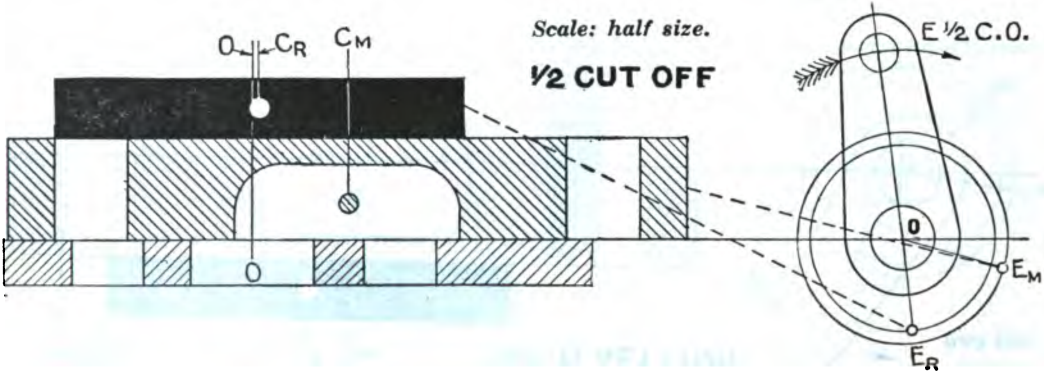

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FIG. 474.—Positions of valves and eccentrics for ½ cut off.

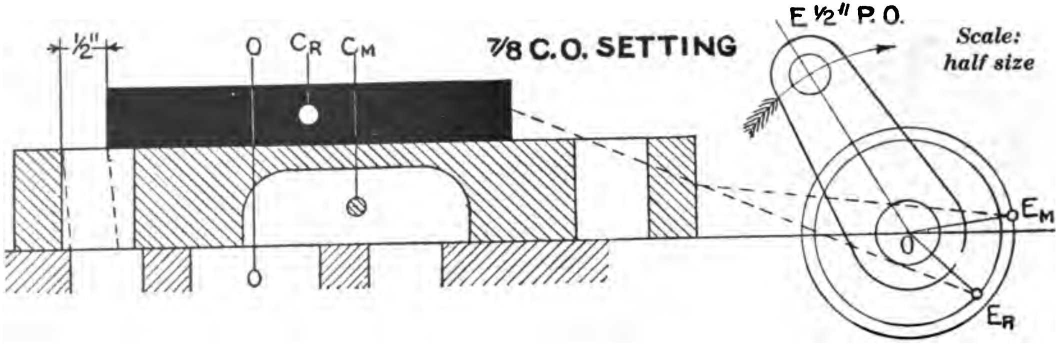


FIG. 475.—Positions of valves, crank, and eccentrics, for ½ in. port opening of the riding valve with ⅞ cut off setting of the riding eccentric.

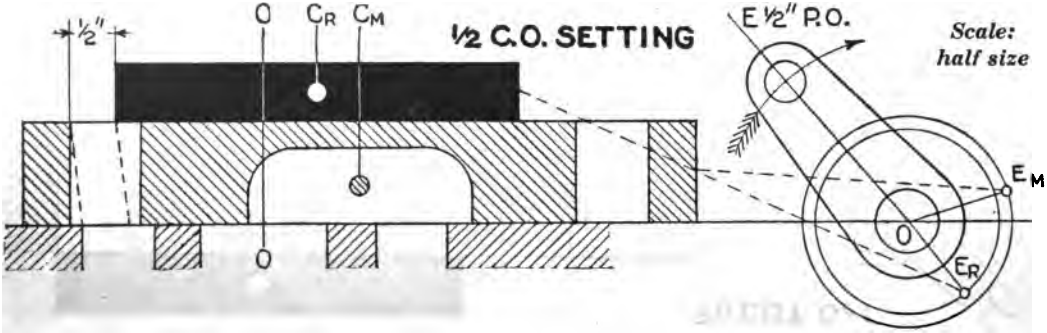


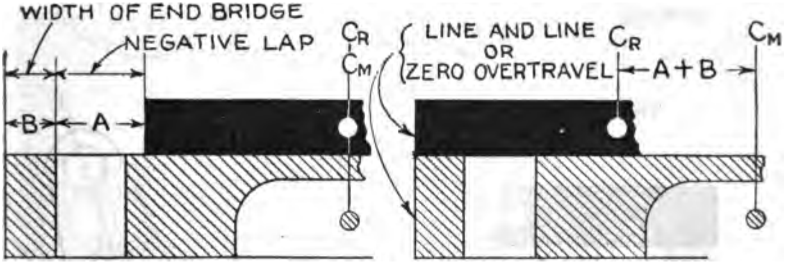
FIG. 476.—Positions of valves, crank, and eccentrics, for ½ in. port opening of the riding valve with ½ cut off setting of the riding eccentric.

easily obtained. For this setting, clearly, the ½ travel of the riding valve on the main valve is equal to *negative lap of the riding valve + width of the end bridge of the main valve* as in fig. 477, the two valves being shown in the position at which over travel begins, that is "line and line" or "zero" over travel position in fig. 478. Hence for this setting, the two eccentric centers, Q and Q′, in the Bilgram diagram will be displaced a distance equal to $C_r\,C_m$, or A + B in figs. 477 and 478. Q Q′, in fig. 479 then is the radius or throw of the *virtual eccentric.*

The cut off corresponding to zero over travel is determined as explained under the figure, being, as indicated in the diagram, fig. 479, ¼ stroke. This leaves no margin for cut off later than the working cut off, and in practice the design should be further modified to increase the range of over travel. A remedy would be to increase the throw of the riding eccentric.

10. Locate the seat limit;

This is done by the same method as explained on page 286, fig. 504, illustrating the seat limit for the Meyer main valve.

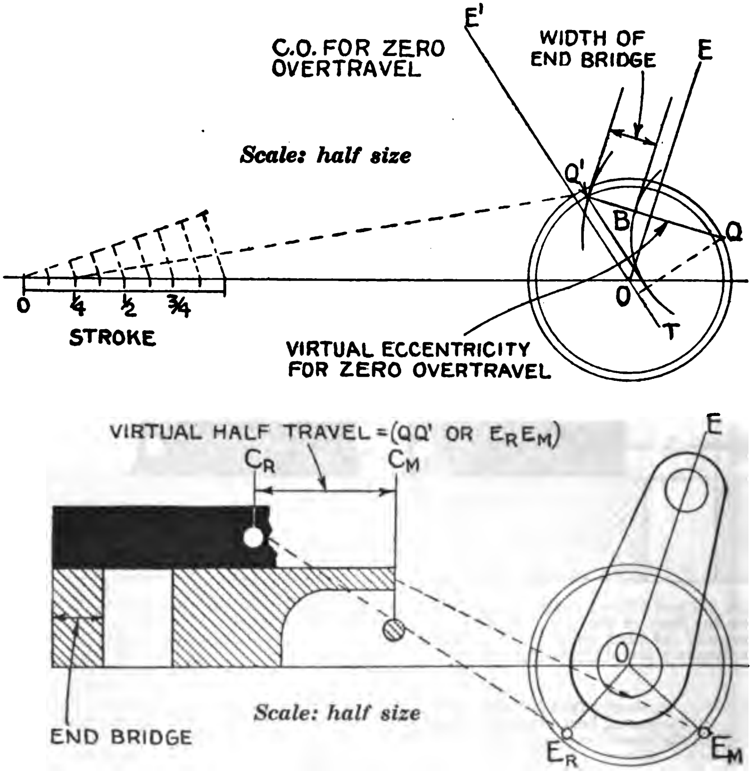


Figs. 477 and 478.—Detail of one end of valves showing that the half travel of the riding valve on the back of the main valve or *virtual half travel* for zero over travel is equal to *riding negative lap + end bridge width.* Fig 477, shows both valves in neutral position, and 478, valves in position of zero over travel.

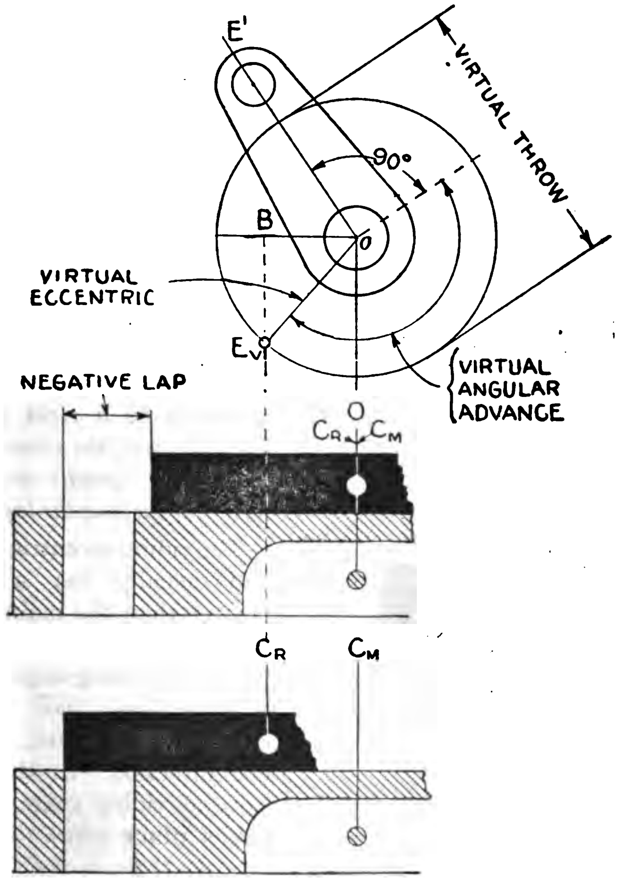
Evidently that portion of the valve which over travels L, is in balance with respect to the steam, hence the shorter the length of the seat the less the load on the valve due to the steam pressing it down on the seat.

Features of Riding Cut Off with Variable Angular Advance.—A study of the example just given will show certain characteristics of the gear which are in brief:

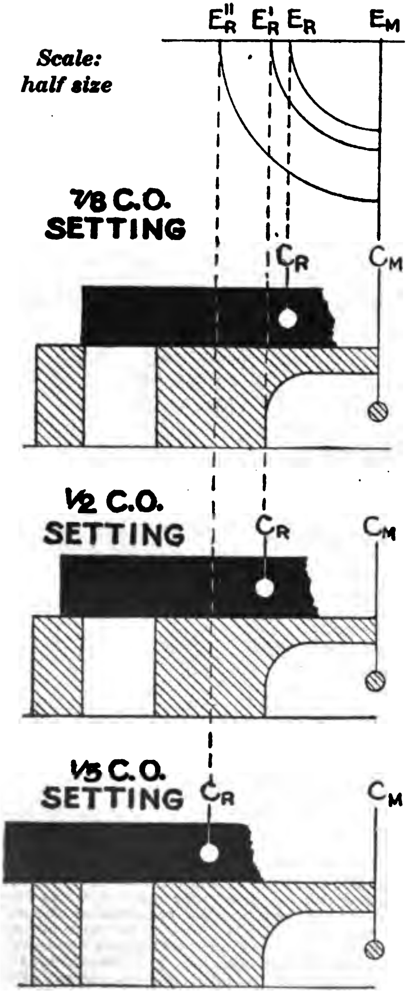
1. *Increasing the angular advance of the riding eccentric shortens the cut off.*
2. *The cut off is "sluggish" for late cut off, increasing in sharpness with the degree of expansion.*
3. *The effective port opening decreases as the cut off is shortened.*
4. *The virtual travel increases as the cut off decreases.*



FIGS. 479 and 480.—Application of Bilgram diagram for *determining cut off setting of riding valve for zero overtravel.* The two travel circles of the main and riding eccentrics, and Q, are transferred from 465, being here drawn on same scale as the valves. With Q, as center and radius equal to A+B, in fig. 477, describe an arc cutting the riding eccentric travel circle at Q′. Q Q′, then is the virtual eccentric for zero overtravel. Draw crank position O E, perpendicular to Q Q′. Evidently Q and Q′, are at their maximum displacement, that is, Q′, is at half travel with respect to Q, hence O E, is crank position when the valve ends are line and line or at zero overtravel, as shown in fig. 480, the distance C_r C_m = Q Q′, being called the *virtual eccentricity*. From fig. 480, *for cut off, the riding valve must move to the right a distance equal to the width of the end bridge.* Hence in fig. 479, on Q Q′, lay off Q′B = width of end bridge, and with Q, as center and radius Q B, describe an arc. Draw through Q′, a tangent to this arc Q′ T, and O E′, parallel to Q′ T. O E′ then is the crank position for cut off corresponding to zero overtravel. Utilizing the main eccentric travel circle as a crank circle, in connection with the dotted line construction to the left, the piston position corresponds to O E′, is found to be ¼ stroke, the design giving overtravel for cut off not later than about ¼ stroke.



Figs. 481 to 483.—The virtual eccentric. By definition: the virtual eccentric is ***an imaginary** eccentric of such throw and angular advance that if keyed to the main shaft and connected with the riding valve, would give it a movement over the main valve* **(regarded as at rest)** *precisely the same as it has when both valves are moving.* Fig. 482 shows both valves in neutral position, and in the diagram fig. 481, O E′, is the crank position of cut off for zero overtravel. With a radius equal to Q Q′ (fig. 479 = C_T C_m, fig. 480) describe a circle whose center is *o*. The diameter of this circle will be the throw of the virtual eccentric, or *virtual throw.* For cut off at *o* E′, evidently the riding valve must move to the left a distance equal to the negative lap. Hence, lay off *o* B=*negative lap,* and project down the dotted line cutting the virtual throw circle at E_V. E_V then is the center of the virtual eccentric, and its eccentricity is equal to *o* E_V, for the setting giving zero overtravel.



Riding Cut Off; Variable Lap.—This method of riding cut off is known as the Meyer gear, and is largely used in marine engines. The riding valve is operated by a fixed eccentric, and the cut off altered by varying the lap of the riding valve.

The gear consists of *a main and riding valve, the latter divided into two plates or blocks connected by a right and left handed screw, the screw serving as a valve spindle and as a means of varying the lap.*

The riding eccentric is fixed and usually has a throw greater than the main eccentric. Its angular advance is 90° for reversing engines, and a little less than 90° for engines which run in only one direction. In this gear, as in all riding gears, **cut off takes place** *when the riding*

FIGS. 484 to 487.—Detail of valve ends with riding valve at end of virtual travel, showing undertravel for ⅞ and ½ cut offs, and overtravel for ⅕ cut off. In fig. 484 imagine E_m as center of main eccentric, fixed in such position that the main valve is in the neutral position, then E_r, E'_r and E''_r are centers of virtual eccentric for the riding valve for throws corresponding to ⅞, ½ and ⅕ cut off respectively. As shown, the virtual eccentrics are at one end of the throw displacing the riding valve a distance equal to ½ the virtual travel.

valve is at a distance from the center of the main valve **equal to its lap,** that is, when the steam edge of the riding valve is line and line with the steam edge of the bridge of the main valve.

The following example will serve to illustrate the features of the Meyer gear.

Example.—Design a Meyer valve gear for an 8 × 10 marine engine suitable for the following conditions of operation: Speed 300 R. P. M.; lead 1/16; cut off range ⅙ to ¾, by riding valve, main valve cut off ⅞; connecting rod ratio 2 : 1.

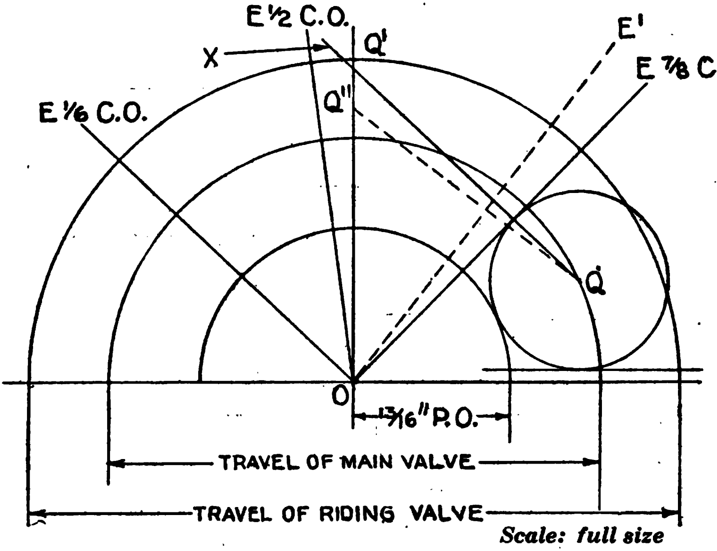


FIG. 488.—Bilgram diagram for main valve of Meyer cut off gear, showing method of locating the riding eccentric to avoid readmission as at Q′. If Q′, be located above Q X, a line drawn perpendicular to latest cut off O E ⅞ C. O., re-admission will not occur.

1. Find area of steam port;

For a steam velocity of 6,000 ft. per minute through steam port.

$$\text{area} = \frac{\text{area piston} \times \text{piston speed}}{6{,}000} \quad \dots\dots\dots\dots \quad (1)$$

$$\text{area piston} = .7854 \times \text{diameter}^2 = .7854 \times 64 = 50.27 \text{ sq. ins.}$$

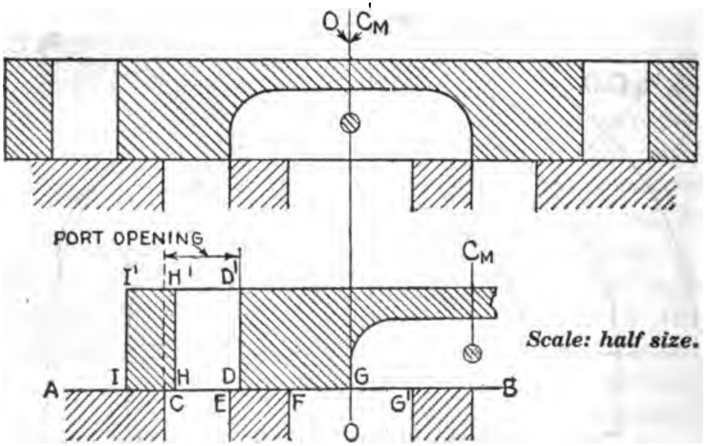
$$\text{piston speed} = 2 \times \frac{10}{12} \times 300 = 500 \text{ ft.}$$

substituting in (1)

$$\text{area} = \frac{50.27 \times 500}{6{,}000} = 4.19 \text{ sq. ins.}$$

2. *Find width of steam port;*

$$\text{width} = \text{area} \div \text{length} \ldots\ldots\ldots\ldots\ldots\ldots \quad (2)$$



Figs. 489 and 490.—Detail of trial main valve and seat for Meyer cut off. On A B, lay off the steam port C E = 11/16 in., the port opening C D = width of port + 1/8 = 13/16 in., and the bridge E F = say 5/8 in. Sketch in valve end in extreme position as in fig. 490. D, will be the steam edge. Lay off D H = width of port = 11/16 in., and draw D D′ and H H′, giving the steam passage through the valve. H H′ = height of exhaust cavity + thickness of metal over cavity = width of port C E + say 3/8 in. = 1 × 1/16 in. The steam passage through valve is usually made same size as steam port so as to reduce friction. I I′, the end of the valve is located far enough beyond H H′, to give a steam tight joint, say 1/2 in. Locate G, the exhaust edge of the valve, so that D G = *outside lap + port* (for zero inside lap) = 15/32 + 11/16 = 1 5/32. The edge G′, of the bridge is so located that G G′ = C E. The center line O O, is now drawn half way between F and G′. The center line C_m, of the valve is now located at a distance to the right of O O = *one-half travel cf main valve*, as measured from the diagram, fig. 488. Fig. 489 shows the valve and seat complete with valve in neutral position. The method of finding the seat limit is later explained.

Call length .8 of cylinder diameter = .8 × 8 = 6.4, say, 6.5 ins., then substituting in (2)

$$\text{width} = 4.19 \div 6.5 = .64, \text{ say } .69 \text{ or } \tfrac{11}{16}.$$

3. *Design main valve*;

The crank position for ⅞ cut off and Bilgram diagram are constructed in the usual way from the given data as in fig. 488 and the main valve shown in detail in figs. 489 and 490. The port opening is made a little larger than the port (⅛ in.) so that the effective port opening at short cut off will not be too small.

4. *Determine travel of riding eccentric*;

Since the engine must reverse, the angular advance of the riding eccentric is made 90°, hence its center Q′, in fig. 488, will be on the vertical line through O. To guard against re-admission Q′, must be located on or above the line Q X, drawn perpendicular to O E, ⅞ C. O., the latest cut off.

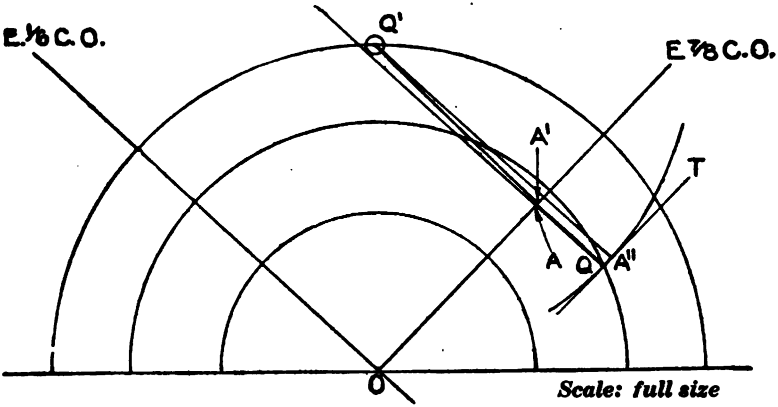


FIG. 491.—Diagram showing laps for various cut offs of Meyer cut off gear.

If Q′, be located on Q X (fig. 488), and the riding valve be given a negative lap equal to Q′ Q, it will cut off and immediately re-admit at crank position O E (⅞ C. O.).

If the center of the riding eccentric be located at Q″, *below* Q X, the riding valve will cut off and immediately admit again when the crank is at O E′, *before* cut off by the main valve, thus disturbing the steam distribution to the cylinder; hence, the importance of correctly locating the riding eccentric, in this case giving it sufficient throw, the angular advance being fixed.

5. *Find lap for earliest ⅙ cut off, and for latest cut off*;

Since both eccentrics are fixed Q and Q′, in fig. 491 remain the same as in fig. 488. In fig. 491, draw crank positions for ⅙ and ⅞ cut offs, and through Q, a line parallel to O E ⅙ C, O.

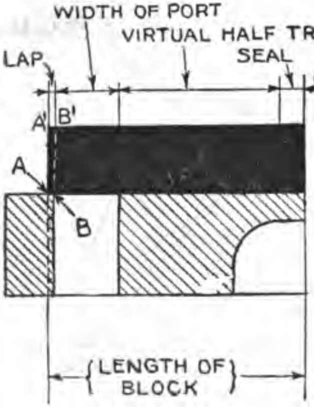
The small lap circle at Q′, tangent to the line parallel O E ⅙ C. O., gives the *positive* lap for ⅙ cut off. For ⅞ cut off, the main valve is displaced a distance Q A, on one side of ⅞ C. O., and the riding valve, a distance Q′ A′. The arc described about Q′, and tangent to a line Q T, through Q, parallel to O E ⅞ C. O., gives the *negative* lap (Q′ A″) for ⅞ cut off.

6. *Determine width of riding valve blocks;*

The blocks evidently **must be wide enough** *so they will not re-admit steam by the back edges when at the end of the virtual travel.*

*For **shortest cut off** the blocks are **farthest apart,*** hence this setting and the virtual half travel must be considered in determining the width. Thus at shortest cut off, width of blocks = *lap + width of port + virtual half travel + seal.*

Thus, fig. 492 shows main valve and one block in neutral position with positive lap A B, for shortest cut off. From steam edge of block, lay off to the right the positive lap A B, and project up to A′ B′. From B′, lay off the width of port, virtual half travel and seal, making the latter say, ¼ inch, giving C, which locates the inner edge of the block.



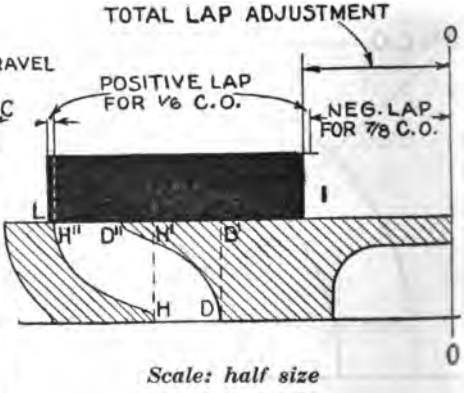


Fig. 492.—Method of finding width of block of Meyer cut off gear.

Fig. 493.—Modified main valve for Meyer cut off gear showing new location of the steam port H″ D″ in top of valve to permit lap adjustment of blocks for latest cut off.

7. *Modify steam passage of main valve to permit lap adjustment of blocks;*

In fig. 493 first draw the block of length just found. From the inner end I, of block lay off positive lap for ⅙ cut off and negative lap for ⅞ cut off, as shown, thus locating the center line O O, of the riding valve for ⅙ cut off setting.

Draw main valve referred to O O, and locate edges H D, of the steam passage through the main valve. The steam passage instead of being straight and terminating at H′ D′, as in the preliminary design, must be curved outward and terminate at H″ D″, to permit negative lap adjustment for late cut off. The point H″, is located at a distance from the steam edge L, of the block equal to the positive lap of earliest cut off.

8. Determine characteristics of the gear;

Construct diagram, fig. 494, showing valve displacements for ¾, ½ and ⅙ cut offs, and diagram, fig. 495, showing valve displacements at middle of admission periods for ¾, ½ and ⅙ cut off settings. From these diagrams one half section of valves are drawn in position corresponding to the cut offs, and mid-admission positions respectively as in figs. 496 to 498.

The figures show that the sharpness or rapidity of cut off increases as the cut off is shortened, the valves moving in the same direction for ¾ and ½ cut off, and in opposite directions for ⅙ cut off.

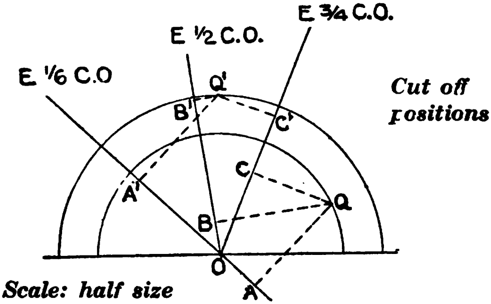


FIG. 494.—Valve displacement diagram for ¾, ½ and ⅙ cut off.

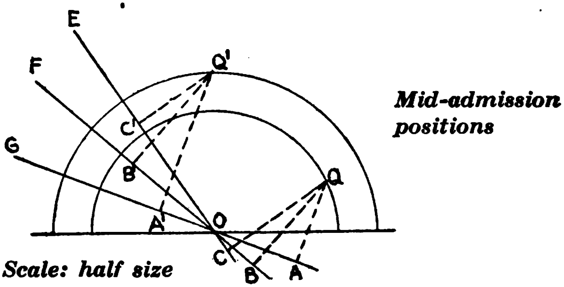
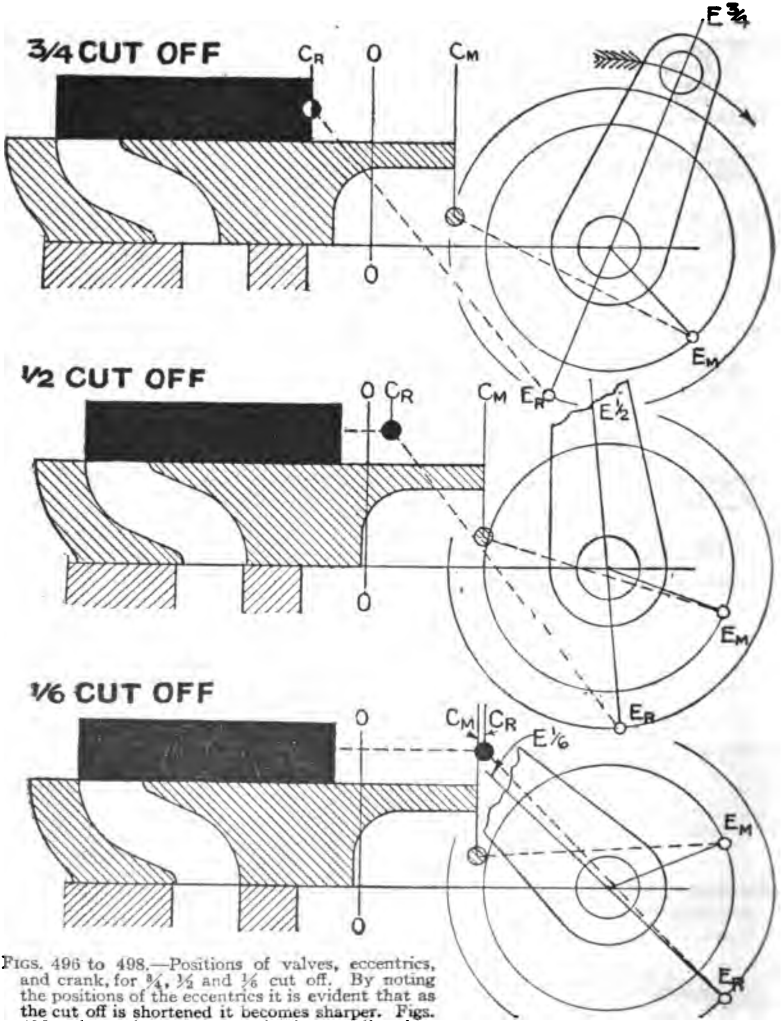


FIG. 495.—Valve displacement diagram for mid-admission crank positions O E, O F, and O G corresponding to ¾, ½ and ⅙ cut off respectively.

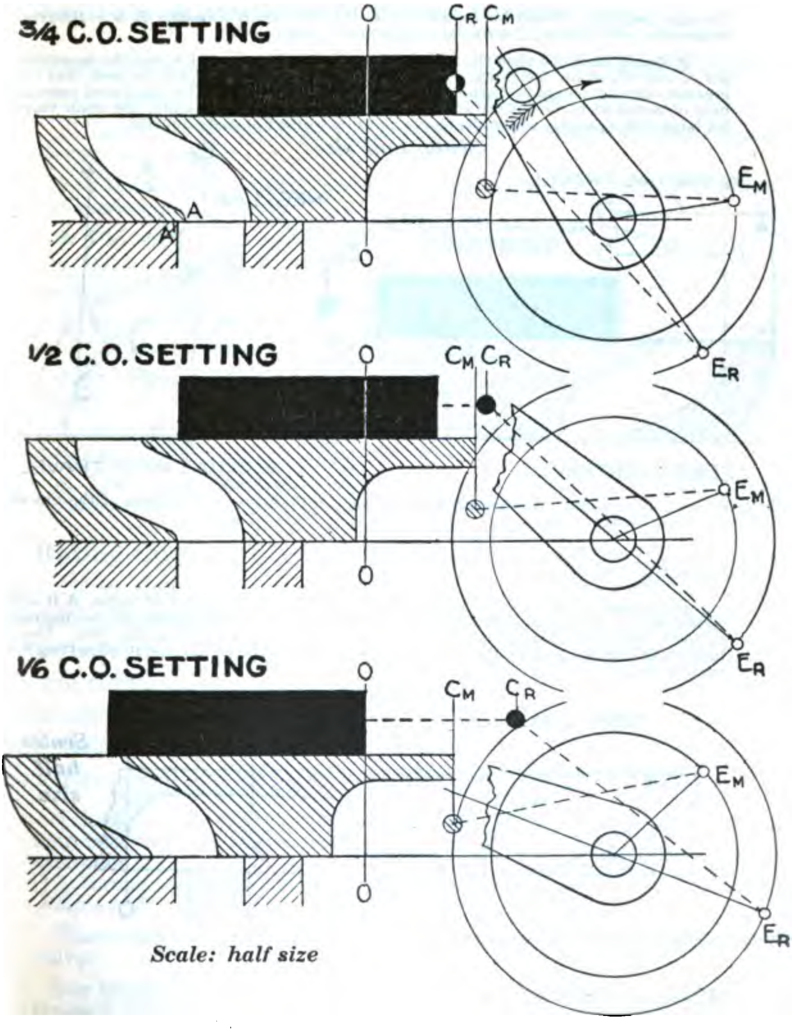
Figs. 499 to 501 indicate that for mid-admission positions, the effective port opening is greater for the ½ cut off setting than for either the ¾ or ⅙ cut off settings.

The reason the port is not fully open in fig. 499 is because the port opening of the main valve exceeds the width of the port. Hence in such case the steam passage



FIGS. 496 to 498.—Positions of valves, eccentrics, and crank, for ¾, ½ and ⅙ cut off. By noting the positions of the eccentrics it is evident that as the cut off is shortened it becomes sharper. **Figs. 496 and 497 show valves moving in same direction, and fig. 498, valves moving in opposite directions.**

Scale: half size



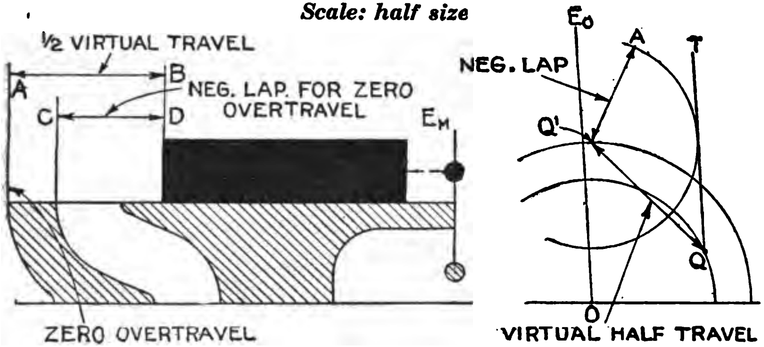
Scale: half size

Figs. 499 to 501.—Positions of valves, eccentrics and crank for mid-admission corresponding to ¾, ½ and ⅙ cut offs, showing effective port opening at these positions.

through the valve should be widened at A (fig. 499) to A′, making A A′ = *difference between the port opening of main valve and width of port in valve.*

It should be noted that the effective port opening in fig. 501 is not the maximum for 1/6 cut off, as by observing the positions of the eccentrics, it will be seen that the greatest opening occurs just after the mid-admission position. The small port opening here obtained at early cut off will indicate the necessity of designing the main valve for large port openings where the engine is to be worked at early cut offs.

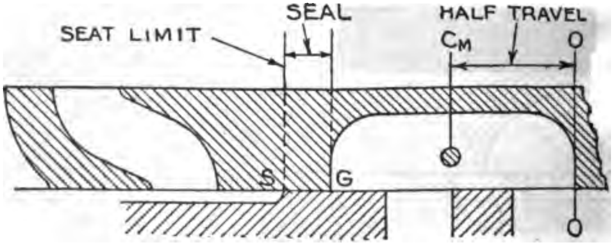
Scale: half size



FIGS. 502 and 503.—Detail of valve end and diagram for finding cut off setting of the blocks corresponding to zero overtravel.

9. Test for overtravel;

Draw one end of main valve as in fig. 502. Lay off, from end of valve, A B = ½ virtual travel, then C D, is the negative lap setting for zero overtravel. In the diagram fig. 503, describe the negative lap circle with radius Q′ A = C D, in fig. 502. Draw tangent Q T, and crank position O E_O, parallel to Q T; then O E_O, is cut off setting for zero overtravel.



Scale: half size

FIG. 504.—Seat limit for Meyer main valve.

10. Locate seat limit.

Draw end of valve in extreme position or at the end of its travel as in fig. 504. From the exhaust edge G, of the valve, lay off the seal, G S, say ½ inch, giving the point S, which is the seat limit.

Features of Riding Cut Off with Variable Lap.

From the example just given illustrating the design of Meyer gear for a marine engine, it will be noted that:

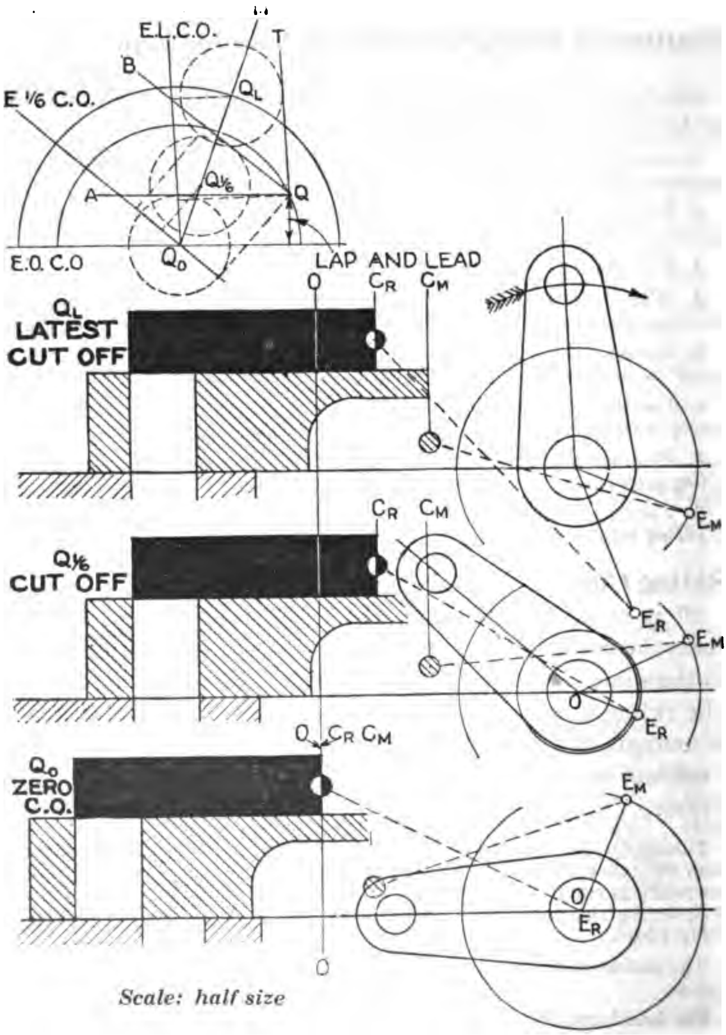
1. *Increasing the lap of the riding valve (that is, moving the blocks apart) shortens the cut off;*
2. *The cut off is "sluggish" for early and late cut off, but somewhat improved for intermediate cut offs;*
3. *The effective port opening decreases as the cut off is shortened;*
4. *Where very early cut off is desired, the main valve should be designed for large port opening, to secure adequate effective port opening at early cut off;*
5. *For reversing engines, the angular advance of the riding eccentric should be 90° to secure symmetrical distribution for both forward and reverse motions;*
6. *For engines running in only one direction the angular advance of the riding eccentric is usually a little less than 90°;*
7. *Re-admission is avoided by the proper location of the center Q', of the riding eccentric;*
8. *The length of main valve may be reduced by shortening the latest cut off of riding valve.*

Riding Cut Off; Variable Travel.—This method of variable cut off is shown in fig. 505 to 512. For illustration and comparison the main valve of the Meyer gear shown in fig. 489 is used with the same travel and angular advance. The maximum throw of the riding eccentric is made the same as for the Meyer gear. The amount of lap to be given the riding valve will depend on the earliest or latest cut off and angular advance of the riding eccentric.

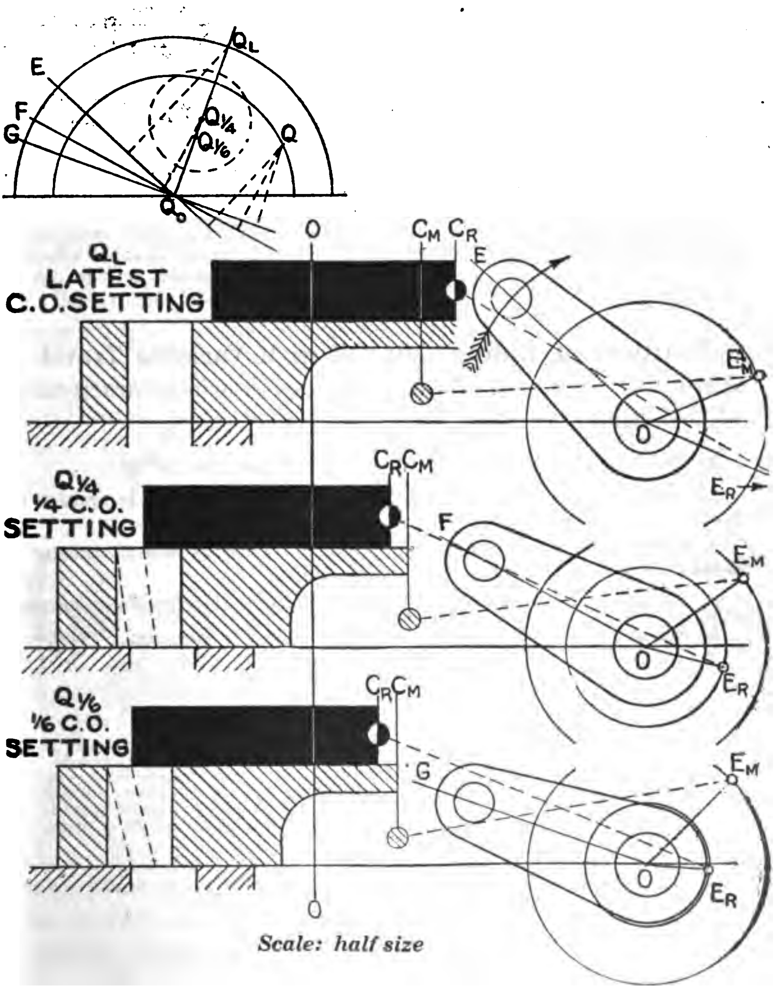
Taking Q_L, fig. 505, for latest cut off with angular advance a little less than 90°, then if the earliest cut off is to be say, zero, the negative lap necessary is equal to the lap plus the lead of the main valve, because a line Q A, through Q, parallel to crank position (E.O.C.O.) is above at a distance $=$*lap*$+$*lead*.

The radius of the negative lap circle thus is equal to *lap*$+$*lead* of main valve.

For latest cut off, describe a negative lap circle through Q_L, and draw tangent QT. A line Q_0 E.L.C.O., parallel to OT, gives crank position for latest cut off.



Figs. 505 to 508.—Bilgram diagram for riding cut off with variable travel and positions of valves, eccentrics and crank for latest, one-sixth, and zero cut offs.



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FIGS. 509 to 512.—Bilgram diagram and positions of valves, eccentrics and crank for mid-admission corresponding to latest, one-sixth and one-fourth cut offs, showing effective port opening. The diagrams show the gradual reduction in port opening as the cut off is shortened, a defect inherent in this type of variable cut off gear.

For ⅙ cut off, draw QB, parallel to OE ⅙ C. O., and describe a second lap circle tangent to QB, and center on line Q_0M, giving the eccentricity $Q_0 Q'_{1/6}$ for ⅙ cut off.

Figs. 505 to 508 show position of valves, eccentrics and crank for latest, ⅙, and zero cut offs. By observing the position of the eccentrics, the quality or sharpness of the cut off may be judged.

In fig. 509 Q_oE, Q_oF and Q_oG, are mid-admission crank positions corresponding to cut off settings of latest, ⅙ and ¼ cut offs respectively.

The valve positions corresponding to O_oE, O_oG, and O_oF, respectively are shown in figs. 510 to 512, from which it will be noted that the effective port opening at mid-admission position rapidly decreases as the cut off is shortened.

Features of Riding Cut Off with Variable Travel.—A study of figs 505 to 512 indicates the following characteristics of this gear:

1. Reducing the travel of the riding valve shortens the cut off;

2. If the range of cut off be up to zero, the negative lap must be equal to lap plus lead of the main valve;

3. For given angular advance and travel of the riding valve, latest cut off depends on the lap of the riding valve;

4. The effective port opening decreases rapidly as the cut off is shortened;

5. Sharpness of the cut off decreases as the cut off is shortened.

CHAPTER 7

MODIFIED SLIDE VALVES

Balanced Slide Valves.—Since the common D slide valve is only adapted to moderate steam pressures, it is necessary where high pressures are used, that there be some means of balancing to prevent excessive friction and wear. This is done by excluding the steam from the top of the valve so that its pressure cannot be exerted in a direction to press the valve against its seat.

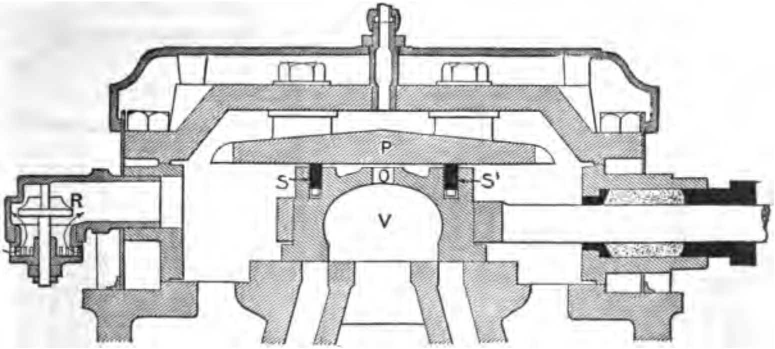


Fig. 513.—The Richardson balanced slide valve. Packing strips S. S', are let into the top of the valve so as to bear against a plate P, thus excluding steam from the top. A hole O, allows any steam which might leak past the packing to escape into the exhaust cavity V. R is a shifting, or relief valve for use on locomotives to admit air into the steam chest, and prevent it being drawn in through the exhaust pipes when steam is shut off, and the action of the piston creates a partial vacuum in the steam chest.

Fig. 513 shows one method of accomplishing this. The top of the valve

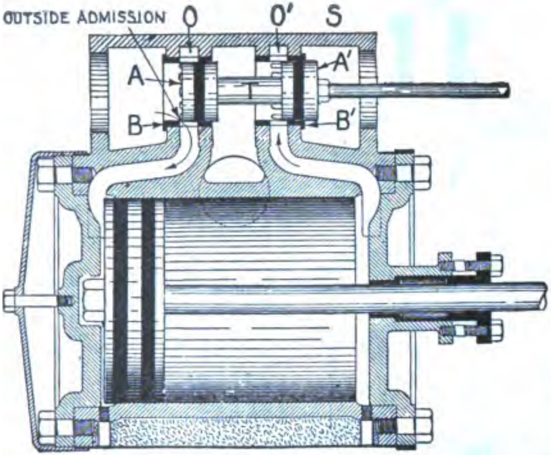


FIG. 514.—***Outside admission*** piston valve. This type consists of two pistons A,A', connected by a central tube T, and working in the barrels, or bushings B,B', which form the seat. O,O', are the ports leading to the ends of the cylinder.

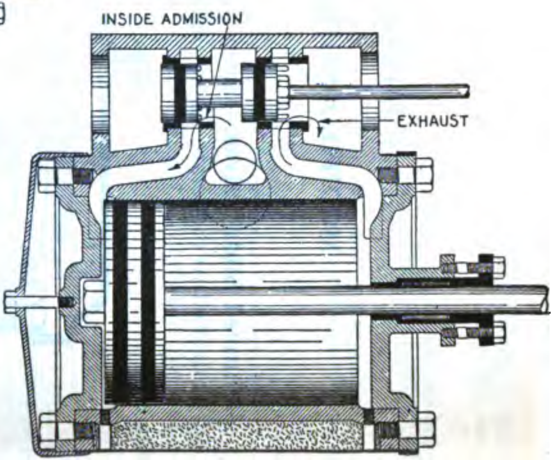


FIG. 515.—***Inside admission*** piston valve. An advantage of this type valve is that the stuffing box is not exposed to the high pressure of the live steam, hence less trouble by leakage; a disadvantage is that, the steam edges of the valves and ports not being visible, the valve is less easily set, the reverse conditions hold for the outside admission valve shown in fig. 514.

NOTE.—In a ***multi-stage expansion engine*** outside and inside admission valves provide convenient receiver connections. Thus in a triple engine the *h.p.* cyl. valve would admit *inside; int.*, outside, and *l.p.*, inside.

is provided with packing strips which bear against a plate P, attached to the steam chest cover, thus making a steam tight joint. The packing is fitted in steam tight grooves, and held in contact with the plate by springs underneath. By this means steam is excluded from the space between the packing. A hole O, allows any steam which may leak past the packing to escape into the exhaust cavity V.

The several methods used in balancing valves will be illustrated in describing the different types.

Piston Valves.—This type of valve consists of two pistons which cover and uncover the ports in precisely the same manner as the laps of the plain slide valve as shown in fig. 514.

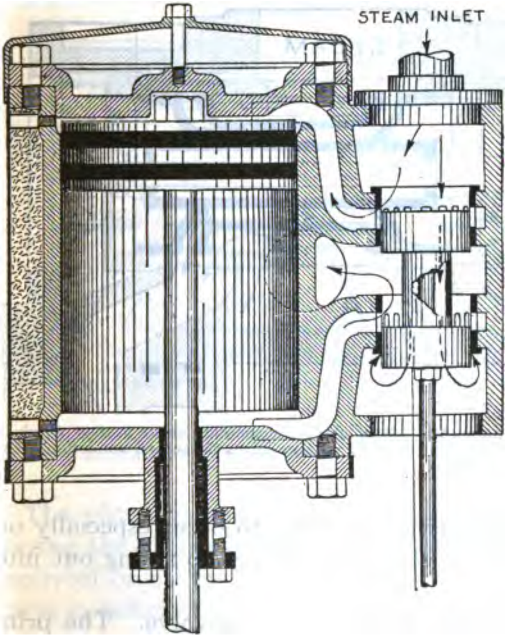


FIG. 516.—**Piston valve with central passage leading to lower port. This type of valve is used where the steam pipe is attached to upper cover and is objectionable in that the central pipe is exposed externally to the exhaust steam which lowers the temperature of the live steam; also the steam pipe must be disconnected to remove the upper valve cover.**

A and A′, are the pistons which are connected by a central tube T. The valve works in the short barrels or bushings B, and B′, which form the seat.

The annular openings O and O′, around the ports form the steam passages leading to the cylinder C. The valve is made to work steam tight by means of the packing rings shown in black.

The barrels are perforated with numerous openings as S, through which the steam passes. The *bridges* thus formed permit the valve to work back and forth across the port without catching or jamming as would otherwise be likely to occur especially on account of the tendency of the packing rings to spring out into the ports.

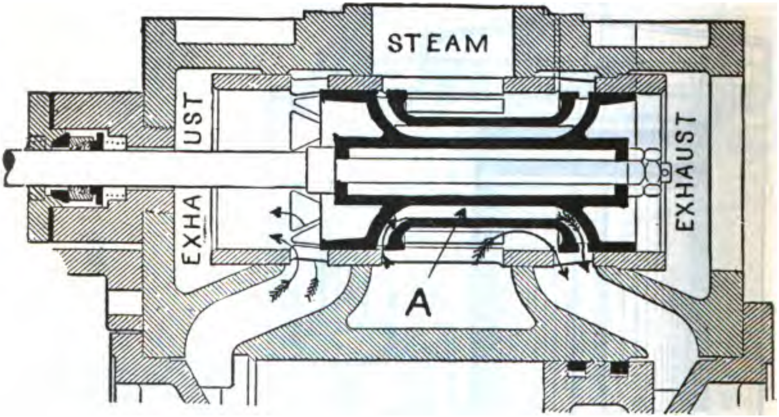


FIG. 517.—Double admission piston valve. An annular supplementary steam passage A, which acts in the same way as the Allen supplementary passage, gives the second admission. The valve is of the inside admission type.

Fig. 517 shows a double admission† piston valve. The principle is similar to that of the Allen valve.

An annular supplementary passage is provided which gives a second admission. Steam is taken from the inside and exhausted at the ends as indicated by the arrows. On account of the surface cut away by the supplementary port, double admission piston valves are seldom provided with packing rings.

The Armington and Sims valve is of the double *inside** admission piston type; instead of the annular passage as in fig. 517.

Steam passes through a central passage, being admitted at the inside and exhausted at the ends.

A double ported† piston valve as used on the Ide engine is shown in fig. 518.

This is also of the inside admission type. As shown in the figure, steam is admitted from the central cavity to the cylinder through the ports B and C, and exhausted at the end through the valve, the course of the steam being indicated by the arrows.

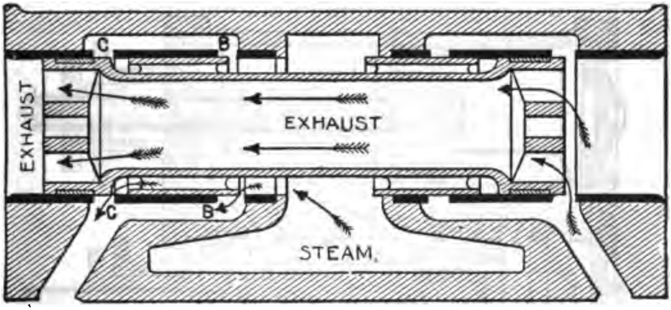


Fig. 518.—The Ide double ported valve giving double admission. but only one opening for exhaust. Admission is from the inside, entering the steam passage through the ports B and C; exhaust passes through the valve to the ends of the cylinder.

The piston valve is especially adapted to compound engines having the pistons working in unison (as on four cylinder locomotives) or having the cranks 180° apart. In either case one valve is sufficient for the two cylinders.

---

*NOTE.—In multi-cylinder engines using high pressure steam this is an advantage since with inside admission for the high pressure cylinder the packing around the valve stem is not exposed to the high initial pressure.

†NOTE.—The difference between a double admission and a double ported valve should be clearly understood. **A double admission valve** *gives two openings to steam both of which lead the steam to a single steam port in the seat, as shown in fig.* 517. **A double ported valve** *gives two openings to steam but a separate port is provided for each*, as B and C, fig. 518.

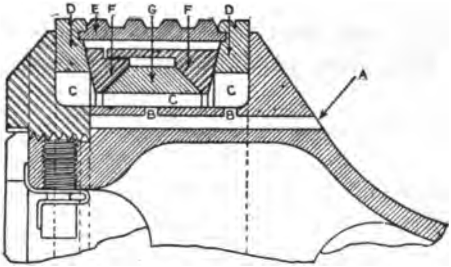


Fig. 519.—Detail of Chandler and Taylor balanced piston valve; A, BB, steam inlet ports to rings; CCC, steam space; DD, snap rings; E, connecting ring; F, F, wall rings; G, wedge ring.

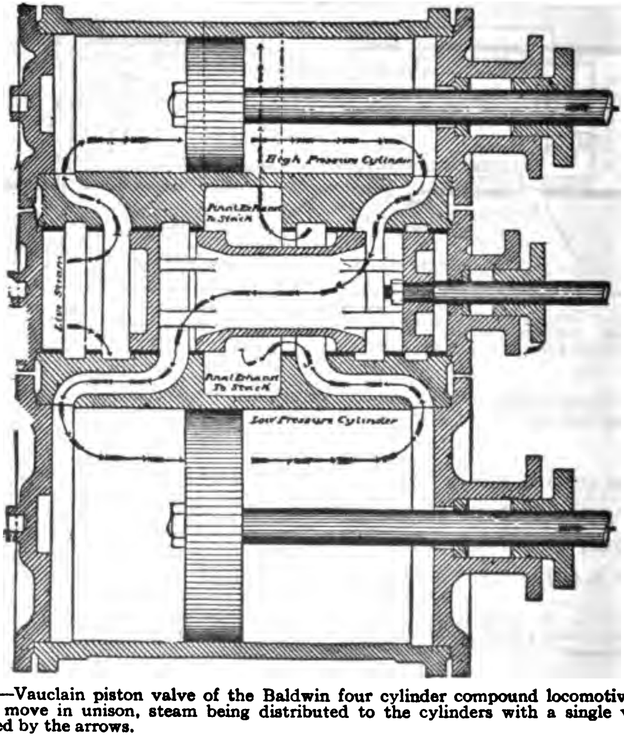


Fig. 520.—Vauclain piston valve of the Baldwin four cylinder compound locomotive. The pistons move in unison, steam being distributed to the cylinders with a single valve as indicated by the arrows.

As applied to locomotives, with pistons working in unison, the arrangement of ports, etc., is shown in fig. 520.

Live steam is admitted to the high pressure cylinder at the ends, and exhausted through an adjacent port in the valve, from which it passes through the valve to an admission port at the opposite end for the low pressure cylinder. The final exhaust passes only through a central depression and passage; the course of the steam through the engine is shown by the arrows.

Fig. 521 illustrates a valve for a compound engine with cranks at 180°.

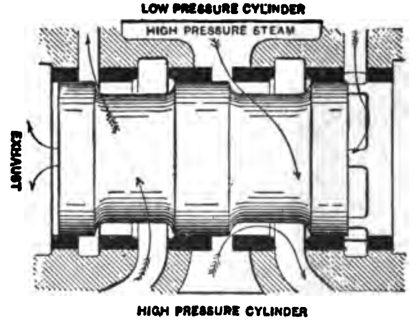


FIG. 521.—Piston valve for compound engine distributing the steam to both cylinders. The cranks being at 180°, one valve suffices for both cylinders. The arrows indicate the path of the steam.

The central part of the valve or seat is surrounded by steam which is admitted through an annular port to the annular valve space, which connects with the high pressure cylinder as shown.

The valve has just opened for steam to the upper end of the high pressure cylinder, and the exhaust from the lower end is just entering the low pressure cylinder, while the low pressure exhaust is escaping from the upper exhaust chamber.

The steam distribution is regulated by five ports: The central port admits and cuts off steam to the high pressure cylinder while the exhaust from this cylinder passes through its steam ports to the steam ports of the low pressure cylinder located at the ends of the seat. The exhaust from the low pressure cylinder is controlled by the outer edges of the valve; at the upper end the exhaust passes through the valve as indicated by

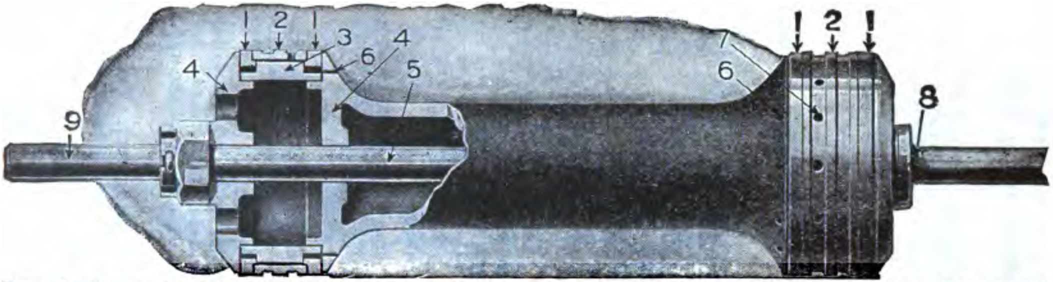


FIG. 522.—Reeves adjustable piston valve. The central portion of the valve, as shown in section, carries in each end a small bronze piston, small in diameter but very long and having water grooves cut as shown. It operates as follows: Steam from steam chest enters through one of the holes, back of piston, on the end of which is fastened a hardened steel plate. This piston is forced out toward the end of the valve against set screw, which is locked in place in the end of the toggle lever which carries a ring shoe on the end. This shoe has two bearings against the inner side of valve ring; the ring therefore, which is made eccentric, is expanded by the action of the piston, the pressure outward being sufficient to overcome the compression pressure in port which tends to collapse the valve ring. Ground joints are made between sides of valve ring and valve body and valve head. The outward pressure on valve ring is 15 per cent. in excess of compression pressure in the port, so that there is 15 per cent. of steam pressure always keeping the valve ring steam tight and in contact with seat, taking up any wear. In case an excessive pressure, such as is caused by a shot of water, the valve ring will collapse and would relieve the cylinder as much as is possible of excessive pressure.

the arrows. This last feature is sometimes utilized on plain piston valves to avoid the separate admission steam passage shown at M, M′, fig. 261.

Fig. 523 illustrates a two valve compound engine with cranks at 180°, and which has a unique system of steam distribution.

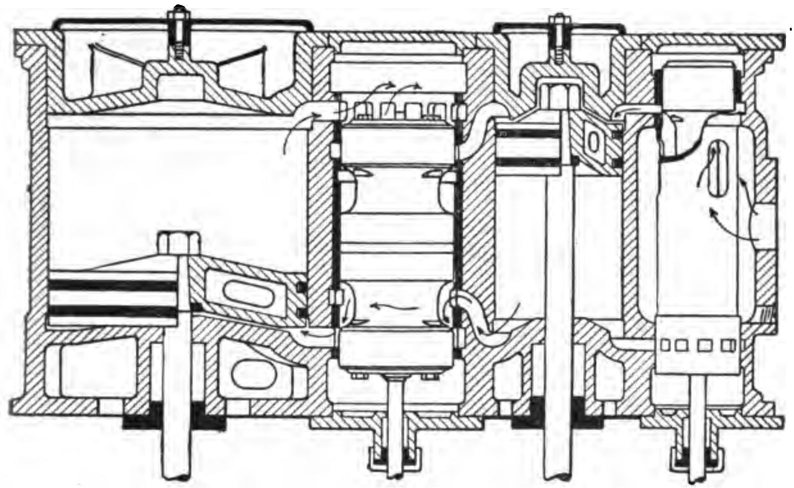


Fig. 523.—Sectional view of the Reeves compound engine with cranks at 180°, showing system of piston valves. The high pressure valve at the right is only for admission, the other valve distributing the steam as exhausted from the high pressure cylinder to the low pressure cylinder. The arrows show the course of the steam in passing through the engine.

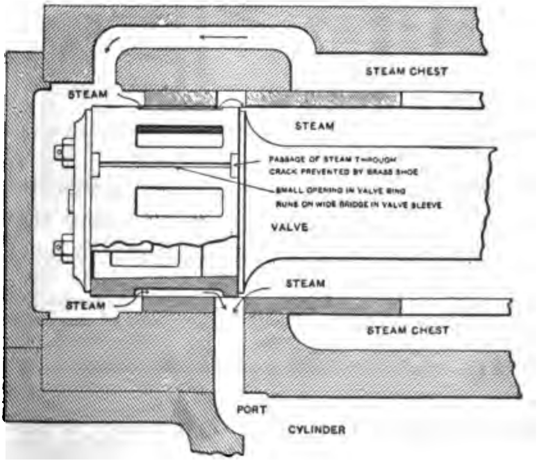

STEAM CHEST
STEAM
STEAM
PASSAGE OF STEAM THROUGH CRACK PREVENTED BY BRASS SHOE
SMALL OPENING IN VALVE RING RUNS ON WIDE BRIDGE IN VALVE SLEEVE
VALVE
STEAM
STEAM
STEAM CHEST
PORT
CYLINDER

Fig. 524.—Reeves double ported adjustable admission piston valves.

Steam is admitted to the high pressure cylinder by a piston valve having internal admission and which acts as an admission valve only, being under control of the governor. The exhaust from this cylinder, and the compression of both cylinders are controlled by the main or central valve which is operated by an eccentric with a fixed travel.

It is obvious that any change in travel or cut off of the admission valve will not effect the cut off in the low pressure cylinder, therefore, changes in load and consequent cut off does not cause excessive compression as in the usual type of compound engines.

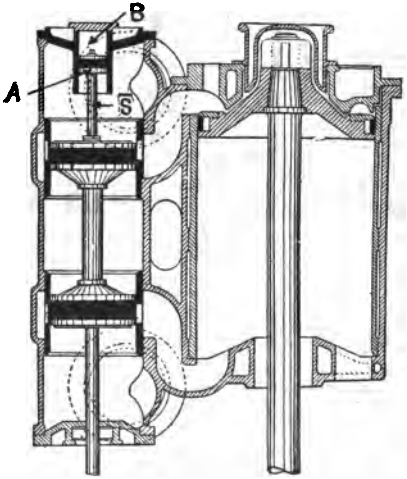


FIG. 525.—Balancing cylinder (B), for balancing the weight of large heavy valves.

On large vertical engines, provision is sometimes made to balance the weight of the valve and thus relieve the valve gear from considerable friction and wear as shown in fig. 525.

An extension S of the valve stem is connected to a small piston A which works steam tight in a cylinder B. The upper end of the balancing cylinder does not admit steam, so that the steam pressure acts upward on the lower face of the small piston and balances the weight of the valve.

The Double Ported Valve.—The difficulty of obtaining sufficient port opening for high speed engines having cylinders of large diameter and short stroke is overcome by providing double steam ports and constructing the valve to open them in unison as shown in fig. 526. It is equivalent to two plain slide valves—a long valve V′, superposed upon a short one V, each having equal steam and exhaust laps.

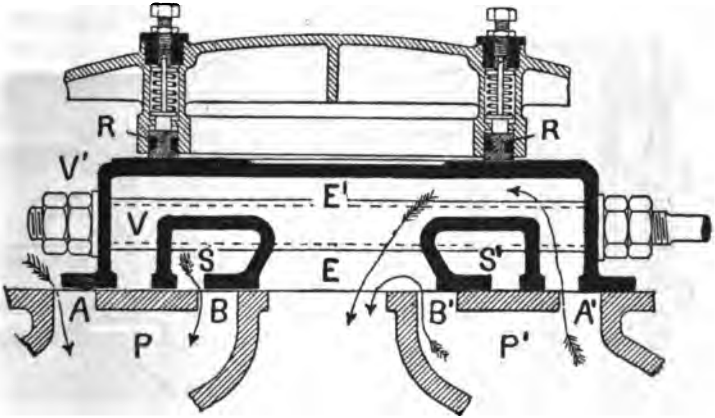


FIG. 526.—Double ported slide valve. There are two openings at each end of the cylinder (A B and A′ B′) for admission and exhaust of steam. The valve is equivalent to two plain slide valves: a long valve V′, superposed upon a short one, and having communicating exhaust passages E and E′.

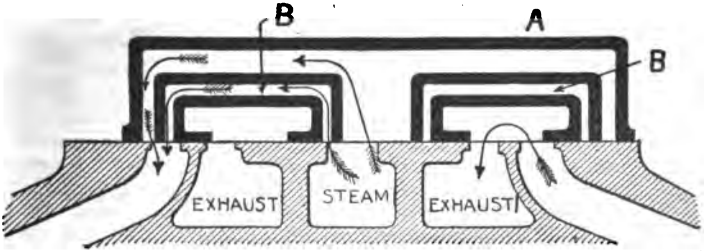


FIG. 527.—Valve of the Russel engine (Giddings type). Steam enters in the center and exhausts through the two adjacent cavities. The action of each end of the valve is similar to that of the Allen valve: the course of the steam is indicated by the arrows. To prevent the live steam lifting the valve from its seat, needle ports (not shown) are used, one connecting the live steam space within the valve to the body of the valve chest, and the second connecting the exhaust with the chest.

The inner valve V, is similar to a plain slide valve except that there is communication between its exhaust space E, and the exhaust space E′, of the outer valve. The two valves form one casting; steam is supplied to the inner valve through the passages S and S′, which communicate with the steam chest at the sides of the valve. Each steam passage to the cylinder has two ports A, B, and A′, B′, and each port is made one-half the width necessary for a single port; hence, the travel is only half that required for a single ported valve having the same area as the port opening. The valve is balanced by means of an equilibrium ring R, fitted to the back of the valve as shown.

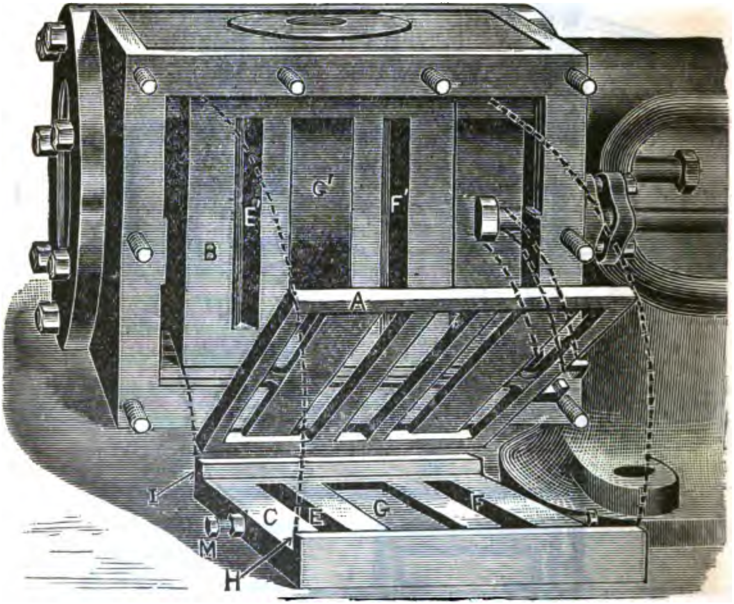


FIG. 528.—Pressure plate valve of the Leffel engine. The object of the plate C, is to relieve the back of the valve A, from the pressure of the steam, this pressure being carried by the two distance pieces H, I, which register with the thickness of the valve. The plate with its depressions E, F, G, forms a second seat, thus making the valve double ported. The dotted lines indicate how the valve and plate are assembled.

Pressure Plate Valves.—Most automatic cut off engines of the high speed type are fitted with valves having two faces, and which provide two, three, and in some cases a greater number of port openings.

The usual construction is shown in fig. 528. The valve A, consists of a long thin rectangular plate which works between the valve seat B, and the pressure plate C. This forms, in fact, a second seat having depressions E, F, G, corresponding to the steam and exhaust ports E′, F′, G′. By means of the rectangular openings in the valve, steam admitted to the ports in the pressure plate passes to the ports in the seat B. The valve is therefore double ported.

The pressure of the steam on the back of the plate is carried by two projecting strips or ***distance pieces*** H, and I, which correspond to the thickness of the valve, thus relieving the latter from the pressure of the steam.

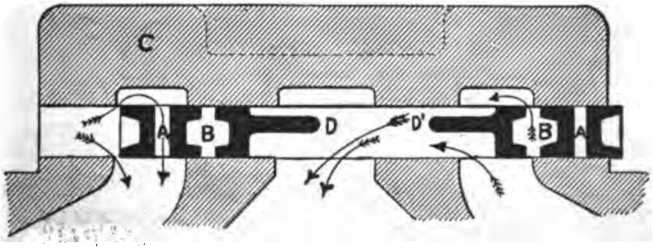


Fig. 529.—The Sweet pressure plate valve. C, is the pressure plate which relieves the valve of the steam pressure. This is a double ported valve with the second admission entering through the passage A. A separate passage B, is used for the second exhaust as seen in exhaust position at the other end of the cylinder.

By means of two adjusting screws M, and N, the ports in the pressure plate are brought opposite those in the seat.

The action of pressure plate valves is best seen from sectional views showing the valve in its lead position as shown in the accompany cuts.

Fig. 529 illustrates the Sweet valve which embodies all the principal features of valves of the pressure plate type.

It is a double face valve, steam being admitted at the extreme ends of the valve, there being two steam edges at each end giving double port opening, as shown by the arrows.

The passage A, conveys steam from the shallow recess in the pressure plate to the main port.

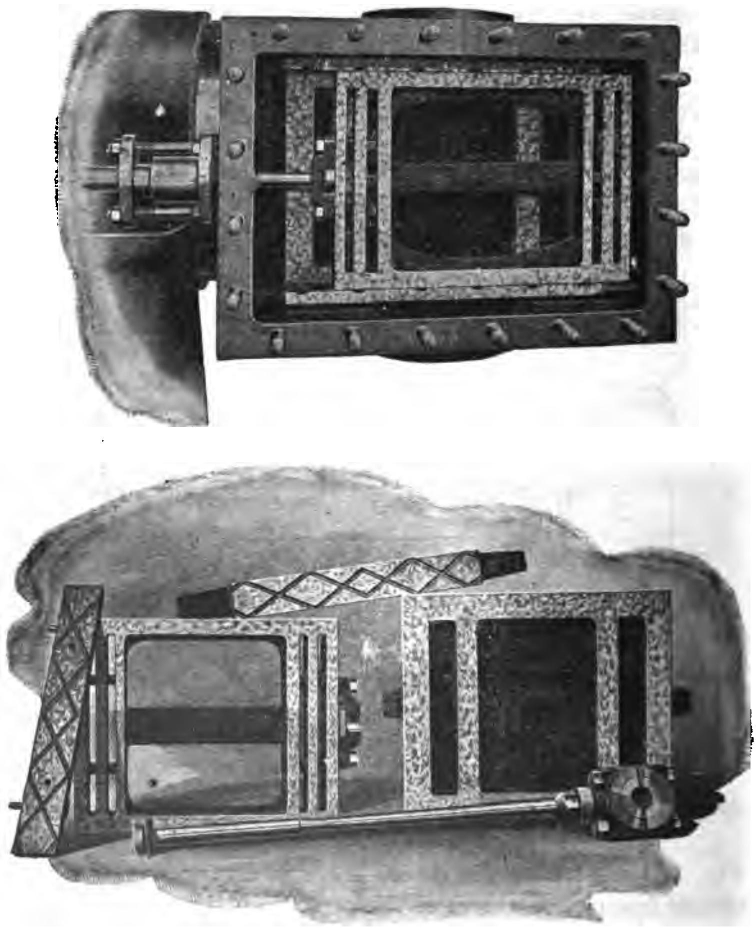


Fig. 530.—Sweet valve and valve stem, showing valve assembled in valve chest of Ames engine, and parts dissembled. The valve with stem, straps and pressure plate as used on the Ames engine. The valve consists of a rectangular casting accurately finished to exact thickness; it operates between the seat and pressure plate which is maintained at the proper distance from the seat by the two strips of iron. The pressure plate is held in position by two flat springs, so arranged that in case the engines receives a charge of water the pressure plate is forced from its seat, allowing the water to pass directly to the exhaust pipe.

The chief object of the exhaust passage B, is to secure a quick opening and closing of the exhaust, so as to avoid wire drawing. After the exhaust is cut off, part of it is compressed and retained in this space before live steam enters the port. Directly after cut off this steam is allowed to mingle with the expanding steam in the cylinder.

The projections D, D′, are for the purpose of protecting the finished surfaces of the pressure plate from the cutting action of the exhaust steam.

In fig. 531 is shown the Woodbury valve which combines the steam features of the Sweet, and Allen valves, giving four port openings to steam, and two to exhaust.

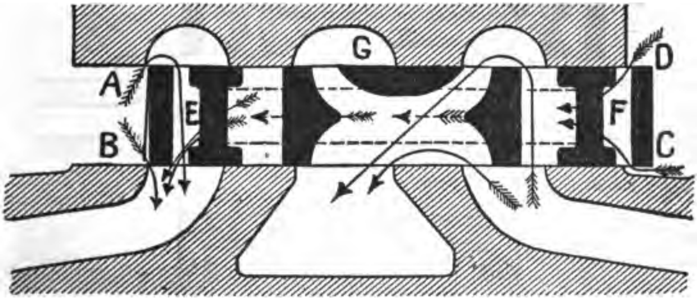


FIG. 531.—The Woodbury pressure plate valve. This valve gives quadruple admission, and double exhaust. The dotted lines show a supplementary passage connecting E and F; this passage acts in the same manner as on the Allen valve.

The openings A and B, act in the same way as those of the Sweet valve.

A supplementary passage is provided along each side of the valve, and connects the steam passages E and F.

This passage is shown by the dotted lines and is similar in its action to the supplementary passage of the Allen valve. The quadruple admission and double exhaust are indicated by the arrows.

A ledge G, is provided as is done in the Sweet valve to protect the finished surface of the pressure plate from the action of the exhaust steam.

A valve which takes steam at the inside instead of at the ends is shown in fig. 532, which illustrates the valve, and used on the Armstrong engine.

This valve gives four openings to admission as indicated in the figure. The steam pressure tends to lift the plate P, and it is therefore held down on its seat by means of the bridle B B.

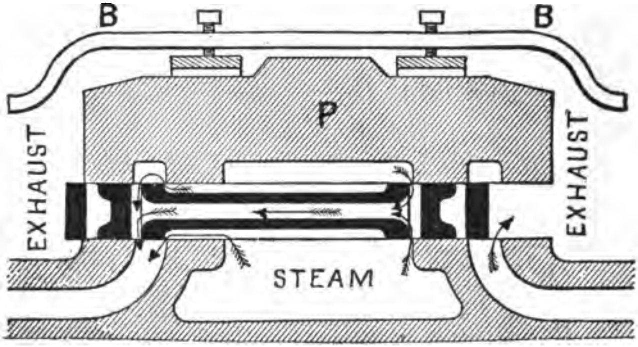


Fig. 532.—The Armstrong pressure plate valve.

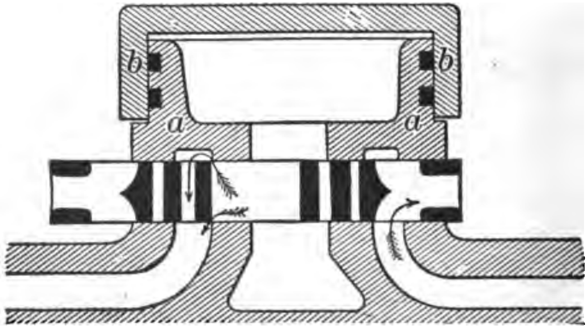


Fig. 533.—The Rice pressure plate valve. Double admission and exhaust, the admission being from the inside.

Another valve taking steam from the inside is the Rice valve, illustrated in fig. 533.

As shown, the valve gives two openings to admission and two to exhaust. The relief plate consists of a piston *aa*, fitted to a cylinder *bb*, which is bolted to the floor of the steam chest. The piston *aa*, bears against distance pieces, and is held in position by the pressure of the steam.

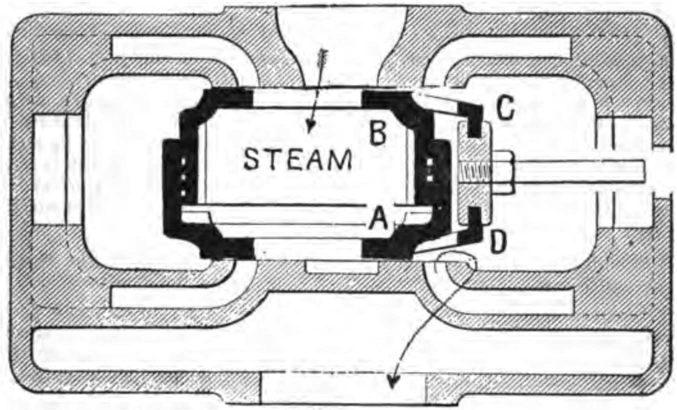
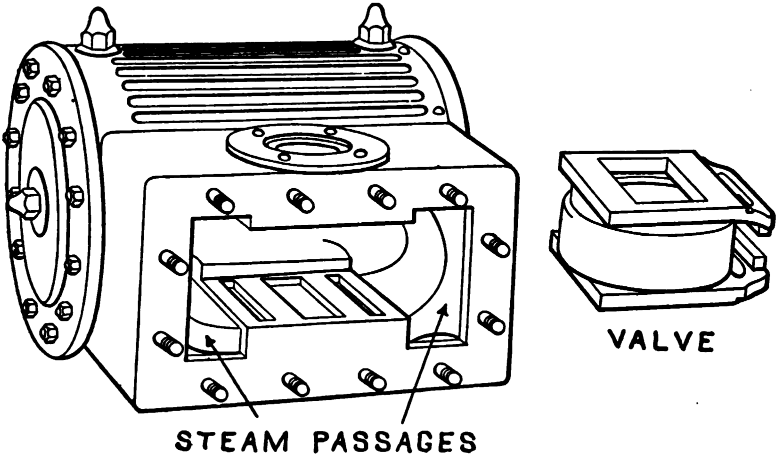


Fig. 534.—The Ball valve consisting of two telescopic cylinders A and B, which are pressed against the seats by pressure of the steam within. A flexible connection is made with the valve stem by the two projecting fingers C and D.



Figs. 535 and 536.—View of the Ball cylinder and valve, showing ports in the lower seat, and the circular exhaust passages. These indirect passages give a rather large clearance; aside from this the valve possesses some good features.

A modification of the telescopic piston and cylinder of the Rice valve is embodied in the design of the Ball valve as illustrated in fig. 534.

This valve consists of two overlapping cylinders A and B, having parallel valve faces at the outer ends. Steam is admitted to the interior of the valves which presses each face against its corresponding seat in the steam chest. The admission is therefore inside, and exhaust outside from which it passes to the exhaust pipe at the bottom as indicated in the figure. The only unbalanced area is that portion of the steam ports which is opposite the cylindrical part of the valve during the exhaust period, the valve being so proportioned as to leave sufficient unbalanced pressure to secure a close contact between the working faces.

A flexible joint is secured with the valve stem by means of the two fingers C, D, which engage in a grove in an end piece attached to the stem.

The valve seats, ports, and steam passages are more clearly shown in fig. 535, and the valve in fig. 536. This valve adjusts itself to wear and has favorable conditions for a permanent, steam tight joint between the two cylindrical parts; it has the disadvantage, however, of a rather large clearance space and indirect steam passages.

CHAPTER 8

REVERSING VALVE GEARS; LOOSE ECCENTRICS

There are many conditions of service where it is frequently necessary to reverse the motion of the engine, as in the operation of locomotives, marine engines, traction engines, etc. Numerous valve gears have been designed by which this is quickly and easily done, moreover in most cases a considerable range of expansion is had by working in the intermediate positions.

The simplest method of reversing an engine consists in *rotating the eccentric around the shaft until it has the proper angular advance for reverse motion.* This operation is shown in figs. 537 and 538. In both figures the crank is shown in its mid position. The corresponding positions of the piston and valve are shown directly above.

In fig. 537, the eccentric is set to run the engine ahead. To reverse the engine the valve must be moved an equal distance to the left of its neutral position so as to admit steam through the right instead of the left port. This is done as shown in fig. 538, the eccentric being rotated through the arc E E′, making the angular advance A′ OE′, the same as A O E. This gives the valve the same linear advance to the left and opens the right port which reverses the motion of the engine.*

A simple application of this principle is shown in figs. 539 and 540.

The eccentric E, is loose on the shaft between a fixed collar G, and a hand wheel H. A stud projecting from the eccentric, and passing through a curved slot in the wheel, can be clamped by a hand nut F.

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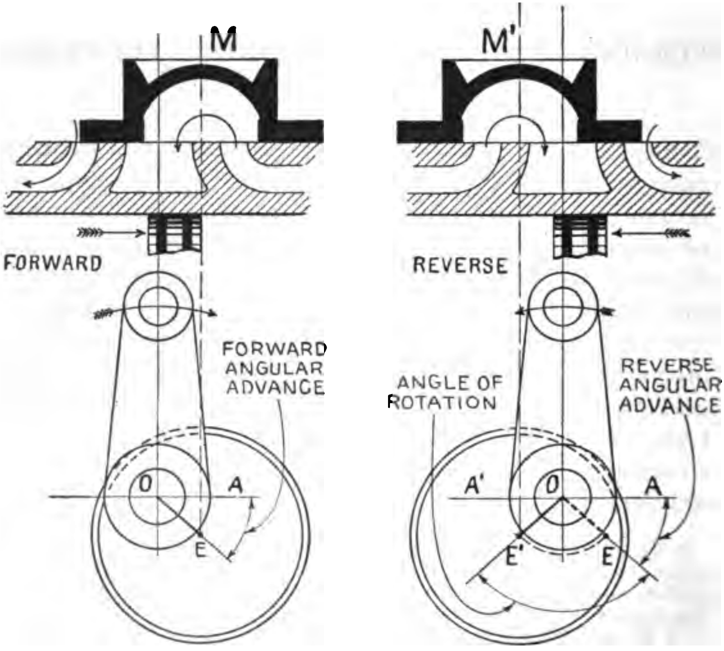
*NOTE.—It should be noted that in the absence of indirect rockers, the eccentric is always placed *in advance* of the crank, that is, ahead with respect to the direction of motion; hence, the direction in which an engine will run is easily determined by noting the eccentric position.

When running forward with the crank at C, the eccentric center is at E, and the nut clamped at F.

To reverse, steam is shut off, and when the engine stops, the nut F, is loosened, then moved to B, and clamped. The length and position of the slot is such that the angular advance A O E = A′ OE′, when the hand nut

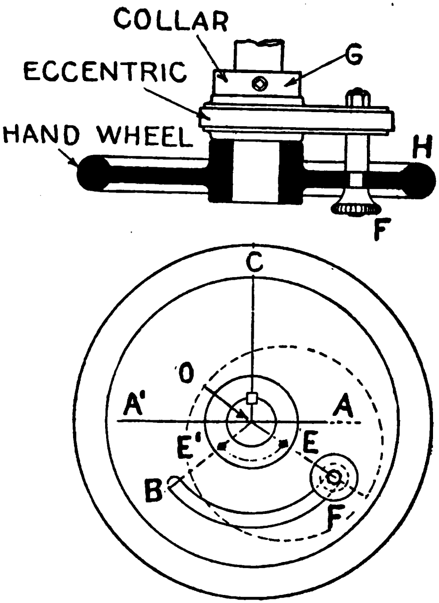


FIGS. 537 and 538.—Simple method of reversing an engine. By rotating the eccentric on the shaft so that it will have a reverse angular advance A′ O E′ (fig. 538), equal to the forward angular advance A O E (fig. 537), the valve will be moved from M to M′, and the engine will run in the reverse direction. The arrows show the steam distribution.

F, is at either extremity of the slot. The letters are the same as in figs. 537 and 538 for comparison.

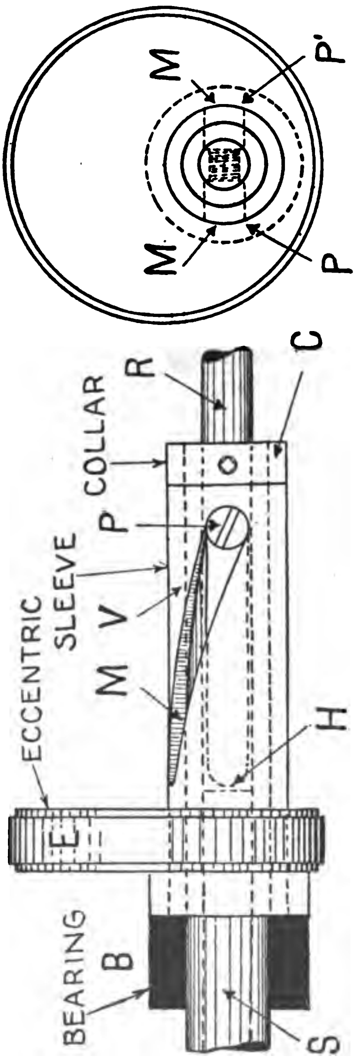
The usual method of rotating the eccentric to reverse on marine engines is shown in figs. 541 and 542.

Figs. 541 and 542 show the construction where the reverse gear is attached to the main shaft. In the figures the eccentric E, is keyed to a sleeve V, which fits so as to easily revolve on the main shaft S; any movement in the direction of the shaft is prevented by the bearing B, and collar C.



Figs. 539 and 540.—Loose reversing eccentric; an application of the principle illustrated in figs. 537 and 538. The eccentric E, is free to turn on the shaft and is held in position by a stud and hand nut F. The stud passes through a circular slot in the wheel, so located that when the stud is clamped at one or the other end, the eccentric is in correct position for forward or reverse motion of the engine.

A spiral slot M, is cut in the sleeve and hole bored in the end of the shaft to H. A straight slot is cut through a portion of the bore from H, to the other end of the spiral slot. The rod R, works in the bore and has attached to its end cross pins P, P′, which pass through the shaft and sleeve slots.



FIGS. 541 and 542.—Loose reversing eccentric; marine type. The pins P, P', attached to the reverse rod R, and passing through the straight and spiral slots of the shaft S, and eccentric sleeve V, respectively, change the position of the eccentric E, when R, is moved to H, thus reversing the motion of the engine.

To change the position of the eccentric, R, is moved, which by the action of the pins in traveling the length of the slots causes the sleeve and eccentric to rotate on the shaft, thus changing the *angular advance.*

By giving the spiral slot the proper pitch, the eccentric may be rotated through the correct arc when P, is moved the length of the slot to reverse the motion of the engine.*

This gear as applied to engines having valves on the side is shown in plan in fig. 543 which illustrates the construction for a compound engine, the cylinder outlines being shown in dotted lines.

The eccentrics E and E', are keyed to a valve shaft S', which is placed directly under the valves and at the sides of the main shaft S.

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*NOTE.—This type of valve gear cannot be used to vary the expansion, because *the travel remains constant, hence the lead becomes excessive for intermediate positions of the gear.*

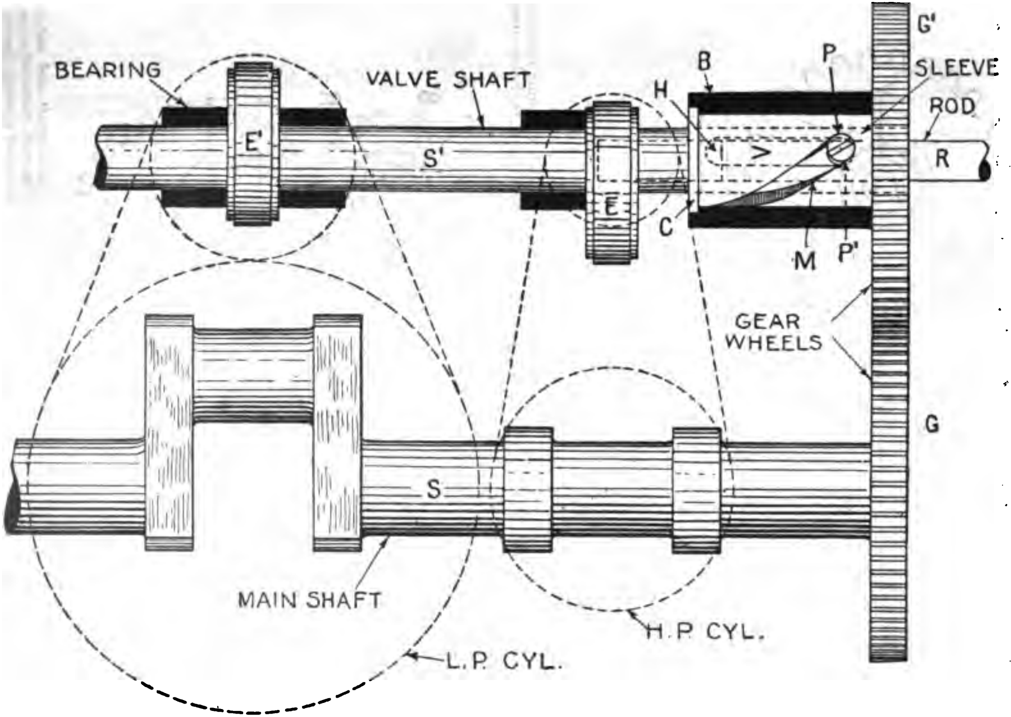


FIG. 543.—Loose reversing eccentric gear: marine type as applied to multi-cylinder engines with valves on the side. The principle of reversing is the same as in figs. 541 and 542, that is, it is accomplished by the action of the pins P, P', attached to the reverse rod R, in moving the length of the spiral and straight slots.

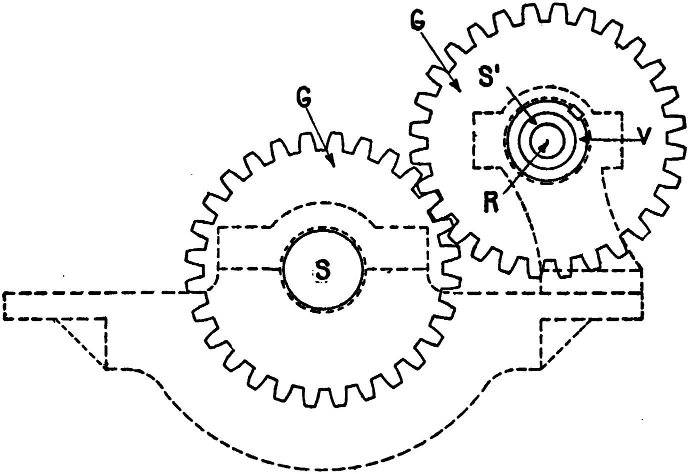


Fig. 544.—End view of fig. 543 showing the two gear wheels G, G′, which transmit motion from the main shafts to the spiral slotted sleeve V, and eccentric shaft S′. R, is the reverse rod.

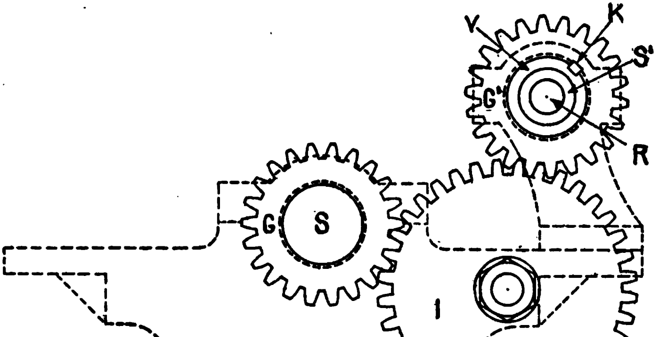
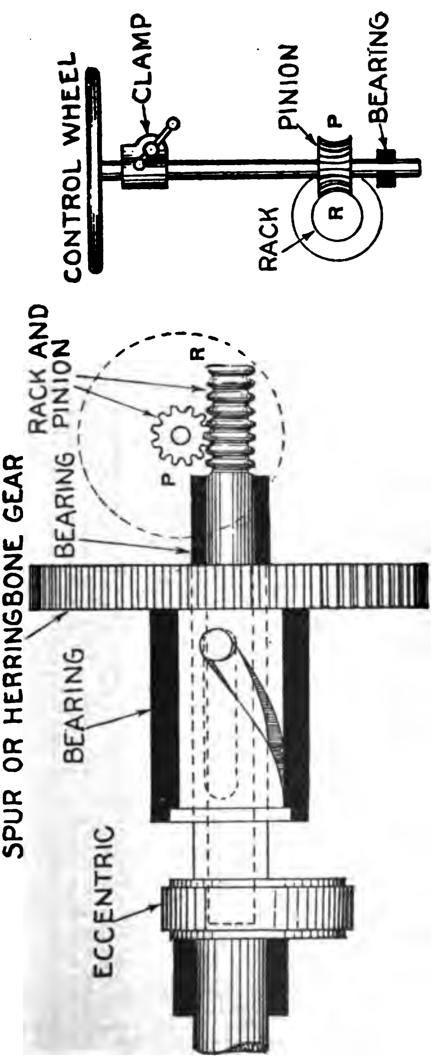


Fig. 545.—End view of loose eccentric reverse gear with idler between main shaft and valve shaft. By using an idler, the diameters of G, and G′, may be made quite small, thus reducing the tangential velocity of the gear wheels which is desirable. V, is the spiral slotted sleeve; S, eccentric shaft, and R, reverse rod. The gear G′, is keyed to the sleeve at K. In design, the sleeve should be thick enough so that it can be firmly keyed.



FIGS. 546 and 547.—A second method of control for loose eccentric reversing gear. In fig. 546, the portion of the reverse gear shown is the same as in fig. 543. *In construction*, a series of teeth are cut in the end of the rod forming a circular rack which engages with the pinion P. The latter is pivoted underneath as shown in fig. 547, and the rod provided with a bearing to hold it in proper engagement with the pinion. Attached to the pinion is a shaft having at its other end a hand wheel. *In operation* the reverse rod R, may be moved to forward or reverse position by the hand wheel and secured in position by the clamp. The construction and operation are clearly shown in the two views.

Motion is transmitted from the main shaft to the small or *valve shaft* S′, by the gear wheels G, G′; these wheels being of the same size, the two shafts revolve at the same rate, but in opposite directions,* the valve movement then, is *indirect*, and the eccentrics are therefore set 180° from the usual positions.

The gear wheel G′, is keyed to the sleeve V, which fits over the valve shaft, and which has a spiral slot M, and a turned projection C, at its end.

This projection or collar and the gear wheel prevent any lengthwise movement of the sleeve as it revolves in the bearing B.

The valve shaft is bored to the point H, and has a straight slot extending from H, to P.

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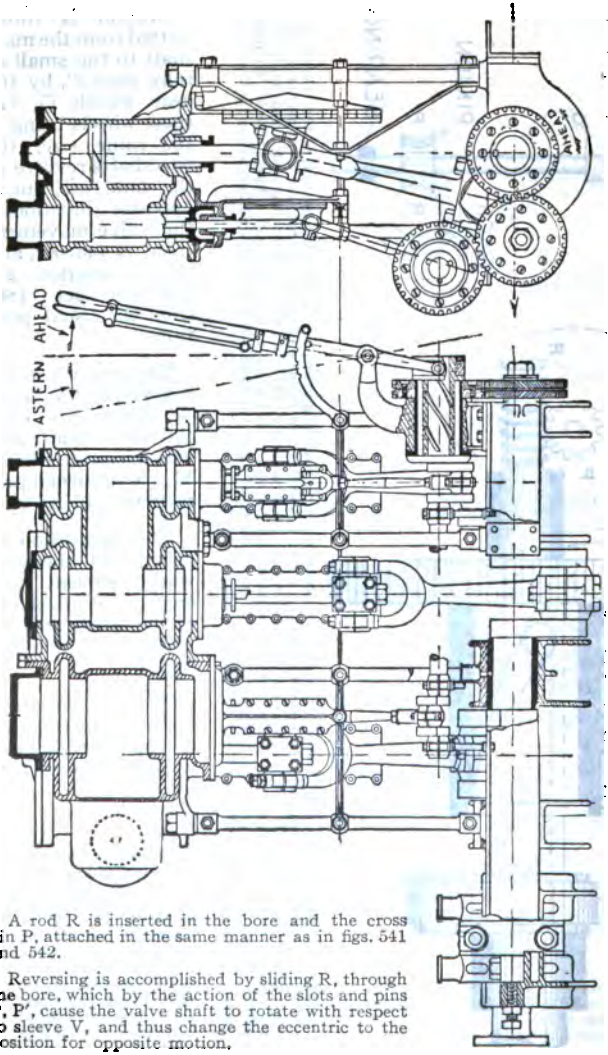
*NOTE.—An end view of the gears G, G′, is shown in fig. 544. Sometimes a third gear or idler is used as in fig. 545. Here the motion of the two shafts are in the same direction, and while there is an extra gear, it has the advantage of reducing the speed at the circumference which is favorable to quiet running.



FIGS. 548 and 549.—Raabe triple expansion marine engine, illustrating *loose eccentric* reverse gear. ***In construction,*** one end of the shaft from which the valves receive their motion is provided with a triple thread of very long pitch, upon which a "nut" is mounted with grooves to fit these threads. The nut is circular in shape and provided with keys, which slide in keyways in a sleeve upon which the gear, which rotates the shaft, is mounted. If the engine be running, the gears, driven by the main shaft will rotate the sleeve with nut and valve shaft as one piece. In throwing the hand lever, the nut is slid outward, and as it cannot rotate inside of the sleeve on account of its keys, the valve shaft has to rotate, changing the position of the cranks, which operate the valve rods, for the astern motion. ***The principle*** is the same as if *an eccentric should be mounted loose upon a shaft, and turned to direct the motion of the engine either astern or ahead,* instead of using two eccentrics for each valve.

A rod R is inserted in the bore and the cross pin P, attached in the same manner as in figs. 541 and 542.

Reversing is accomplished by sliding R, through the bore, which by the action of the slots and pins P, P′, cause the valve shaft to rotate with respect to sleeve V, and thus change the eccentric to the position for opposite motion.

CHAPTER 9

REVERSING VALVE GEARS, LINK MOTIONS

The so-called Stephenson* Link Motion.—This, though one of the earliest forms of reverse gear is probably used more extensively than any other; in the opinion of the author it is as

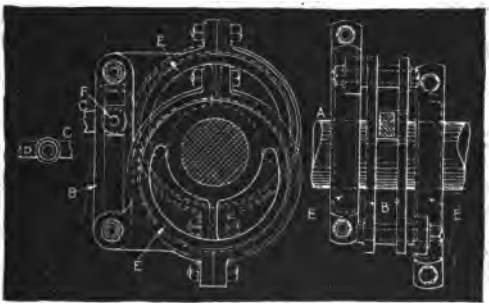


Fig. 550.—***The Williams Link.*** The following quotation from Burgh's Link Motion and Expansion Gears is a full history of the so called Stephenson link motion including conversation of inventor with the author (Burgh). Howe's invention was suggested by the Williams link shown above. "The inventor of the link motion in its simplest original and best form is Mr. W. Howe, who introduced it in the month of August in the year 1842. He was then a working mechanic in the employment of Messrs. Robt. Stephenson & Co., Engineers, Newcastle-on-Tyne. The history of the invention may now be given in Mr. Howe's own language, as expressed to the writer. A species of link motion (shown above) was, just before this date, suggested by Mr. Williams who was a young gentleman apprentice in the works at the time. In the figures, A, indicates the crank shaft B, the proposed link C. is the connecting rod, connecting the link to the valve rod D; E, are two eccentrics, and F, the block for reversing the motion of the valve and engine. It will be easily seen that the suggestion could never have been of the least practical use, because one eccentric bank would displace the other when in motion. Several persons employed in the works saw the drawing Mr. Williams had made, and amongst them Mr. Howe, but no one brought it into a state for practical application until August 1842, when Mr. Howe made a pencil sketch and a rough wooden model of his link motion, and both of the originals are now in the South Kensington Museum. This model so perfectly indicated what the curved link should be, that, acting upon the advice of his friend, Mr. Howe showed it to Mr. Hutchinson, then the manager of Stephenson's Works, who at once saw the worth of its application practically, not one as a reversing but also as an expansion gear for working the slide valve, and he (Hutchinson) sent the model at once to Mr. Robert Stephenson, then in London, who also approved of it immediately he saw it. At the time, Mr. Howe was engaged in making a working model of a wedge motion for two locomotives, being built, but was directed to substitute for this, his link motion. He then made a full size model and proved the adaptation of the link motion for any grade of cut off. ***The dimensions were:*** outside lap one-half; under lap one-sixteenth; port opening 1 in.; throw 3: length of eccentric rods five in.

beautiful a piece of mechanism as ever was invented. The name *shifting* link is sometimes used to distinguish it from the *stationary* or Gooch link.

The Stephenson link was originally intended for reversing only, but within certain limits it is used to advantage as a variable cut off or so called expansion gear. This feature is made use of especially on locomotives, and marine engines.

As shown in fig. 551, it consists of a link L, block M,

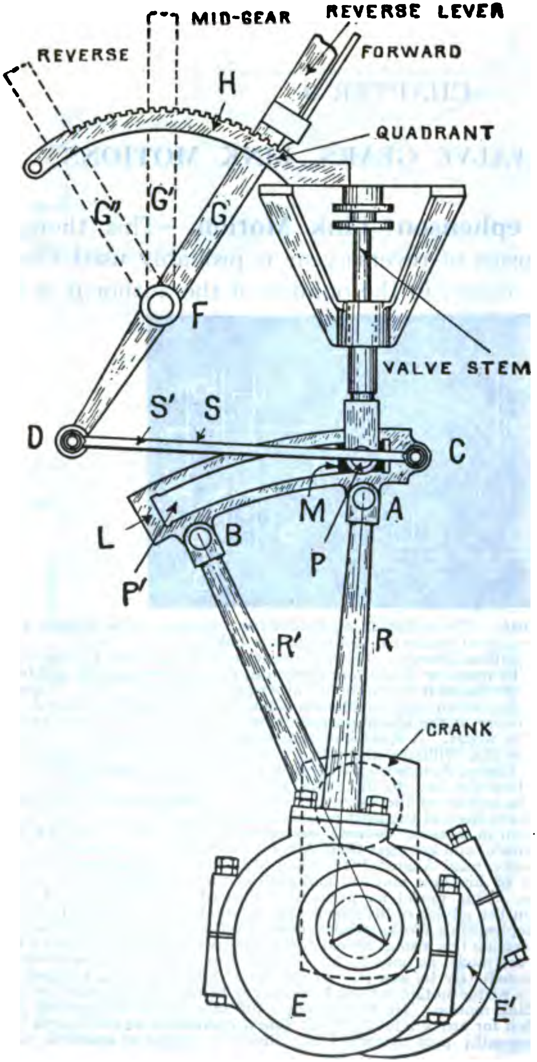


FIG. 551.—The so called Stephenson, or ***shifting link***. There are two eccentrics E and E′, whose rods R, and R′, are connected to the link at A, and B. When A, is opposite the block M, as shown, the engine runs in a forward direction. ***To reverse,*** lever G, is moved to G″, which brings B, opposite the block, thus the motion of the reverse eccentric is imparted to the valve stem, and the engine reversed.

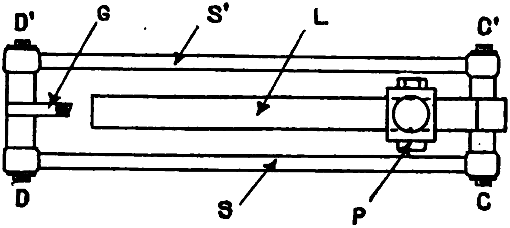


FIG. 552.—Plan of shifting link showing double reach rods S and S′. ***With two rods there is no lateral or twisting strain on the stem in reversing;*** this is a point well worth noting by anyone intending to purchase an engine, the offset form of construction being objectionable. The reach rods are pivoted to the link at C,C′, and to the reverse lever G, at D,D′. P is the valve stem pin.

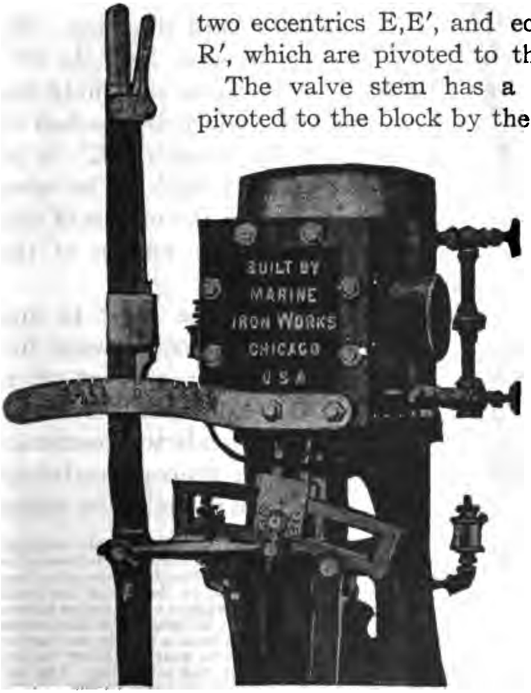


FIG. 553.—Upper end of single cylinder marine engine ***showing link with adjustable block.*** The link is provided with ***double reach rods*** having central connection on each side of the link preventing lateral strain.

two eccentrics E,E′, and eccentric rods R, and R′, which are pivoted to the link at A and B.

The valve stem has a forked end, and is pivoted to the block by the pin P. Reach rods S and S′, (one on each side of the link) connect the latter with a notched quadrant H, and latch I, which retains it in any position.

The link which consists of two curved bars bolted together at the ends, freely slides on the block when the reverse lever is moved, and to a limited extent in operation.

If the block be at one end of the link; the motion of the eccentric attached to that end of the slot is transmitted to the valve; when the block is at some intermediate position, the valve receives the combined motion of the two eccentrics; if the block be at the middle of the slot, or *mid-gear position*, the valve does not admit steam to the cylinder. As shown in the figure, the block is at that end which is attached to the forward eccentric E, hence the engine runs in a forward direction. By moving the reverse lever to G″, the link slides to the right until the other end P′, which is attached to the backward eccentric E′, is in contact with the block. The valve then partakes of the motion of this eccentric and the motion of the engine is reversed.

With the reverse lever in any intermediate position between full gear and mid-gear, the cut off is shortened, because the motion of one eccentric tends to counteract that of the other; the combined effect is to reduce the travel of the valve.

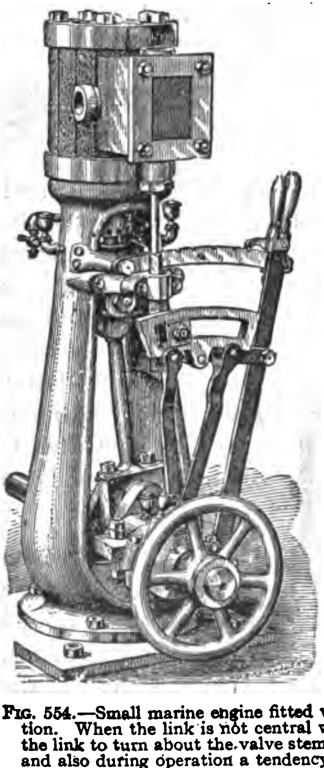
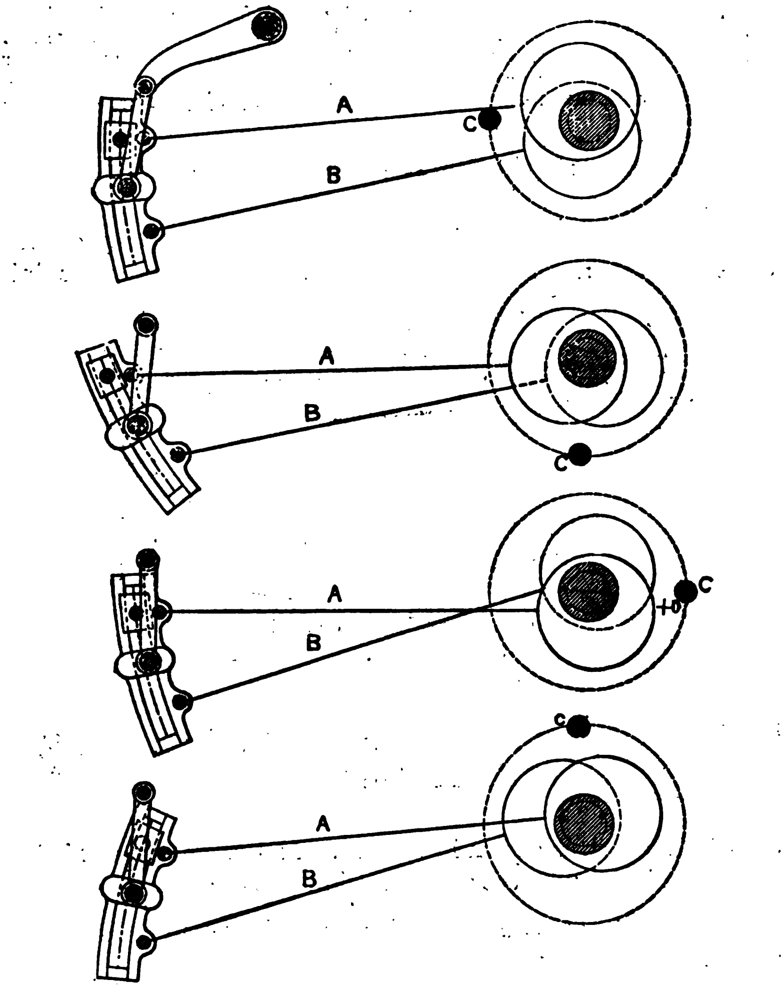


FIG. 554.—Small marine engine fitted with an offset shifting link; an objectionable construction. When the link is not central with the axis of the valve stem there is a tendency for the link to turn about the valve stem axis every time the link is shifted by the reverse lever and also during operation a tendency to turn to and fro is caused by the *slip* of the block. The latter effect is augmented by the objectionable location of the pivot at the center instead of at the end of the link inasmuch as the slip is increased when the pivot is at the center. This turning tendency is resisted by providing the valve stem with a square end section working in a bearing as shown. Evidently any lost motion due to wear will allow the link to get out of alignment and sometimes cause it to work roughly or stick in shifting. The only advantage due to offsetting the link is that it allows more room for a main bearing. In the above example, the square section of the valve stem should be much larger and preferably shaped as a flat bar of a width considerably greater than its thickness. The bearing should be adjustable for wear.

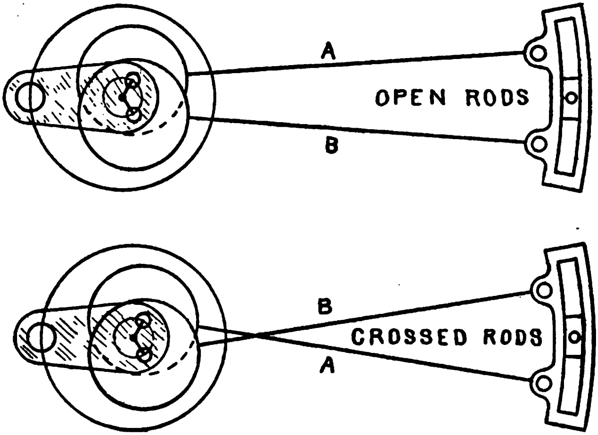


Figs. 555 to 558.—Movement of the link during one revolution. The figures show the positions of the link gear when the crank C is on the dead center, and at ¼, ½ and ¾ of a revolution. The point of suspension being at the center, as on locomotives, the slip is considerable.

Some of the different positions taken by a link for one revolution of the crank are shown in figs. 555 to 558, which illustrates a locomotive link motion in full gear for four positions of the crank. Fig. 555 shows the gear with crank C at the beginning of the stroke; the other figures illustrate the position of the link when the crank has made ¼, ½, and ¾ of a revolution.*

Ques. What is the difference between open and crossed rods?

Ans. The eccentric rods are said to be open, if they do not cross each other, when the eccentric centers lie between the link and shaft center as shown in fig. 559. If the reverse condition obtain, as in fig. 560, the rods are said to be crossed.



FIGS. 559 and 560.—Diagrams illustrating open, and crossed rods. In shortening the cut off by "hooking up," **open rods give increasing lead, crossed rods, decreasing lead. When it is intended to work the engine linked up, as with a locomotive, it is advisable to have the rods open, as a greater range of expansion is obtainable with less reduction of port opening than with crossed rods.**

Ques. What is the effect of open and crossed rods on the steam distribution?

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*NOTE.—In figs. 555 to 558 the reach rod is shown attached to the central portion of the link instead of at the end. This construction is for locomotives on account of the position of the rocker arm but the action of the link is not so good as when the attachment is at the end as in figs. 575 and 576.

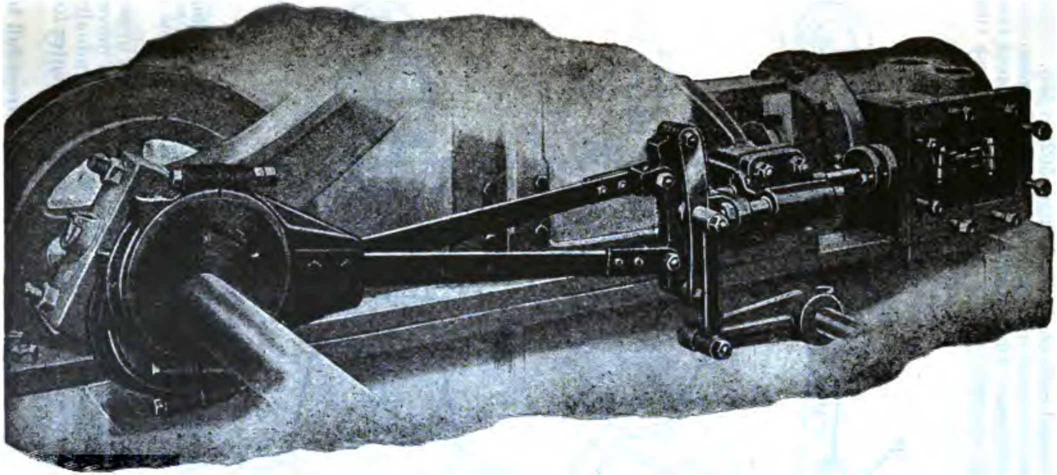
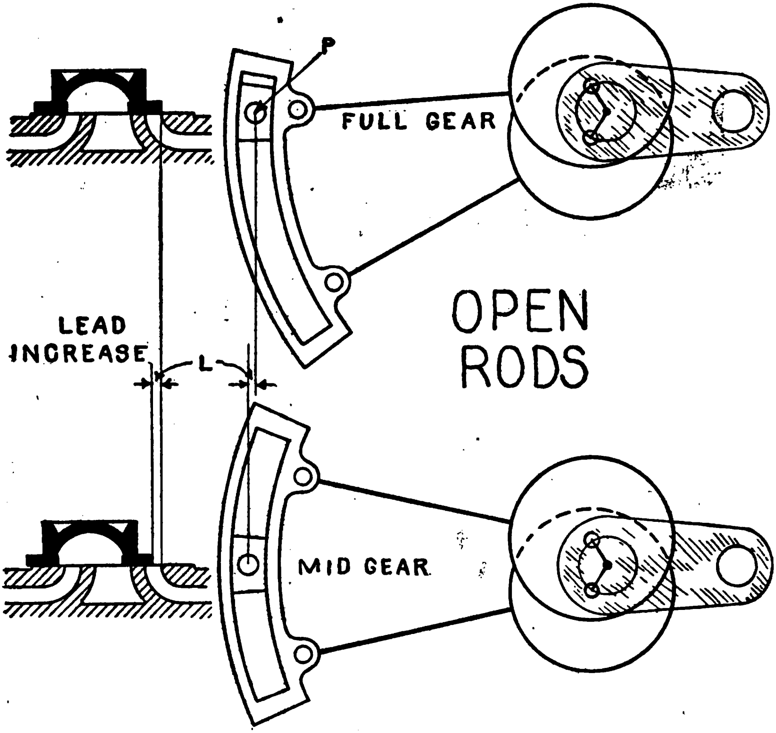


Fig. 561.—Link motion as applied to the Lidgerwood double cylinder hoisting engine; type used on steam lighters, barges, etc. The links are operated by a hand lever through the medium of a rock shaft and suitable connections. The moving parts of the link are counterbalanced. The link jaw and connection to the arm of the rock shaft are of bronze, the link being of cast iron, and the block of steel. The links are slotted in a special machine which cuts the slots to exactly the required radius and the link blocks are made as a steel ring of the required radius, turned to size and cut into suitable lengths. The parts are then fitted perfectly by hand scraping. The link jaw is formed with guides for the link, and it also extends sufficiently to receive the end of the link jaw support stem which moves backward and forward with the valve stem in a babbitted bearing. This gives a wide support to the whole link mechanism which prevents any tendency on its part to twist out of line. The reverse lever is provided with thumb latches and a quadrant. A central notch in the quadrant holds it in the off position, and notches near the outer ends give the engine steam at full stroke in either direction. Other notches between provide for a cut off, usually at about five-eighths stroke.

Ans. With open rods the earlier the cut off, the greater the lead; with crossed rods the lead decreases with the shortening of the cut off. In either case the port opening is reduced, but to a lesser extent with open rods.

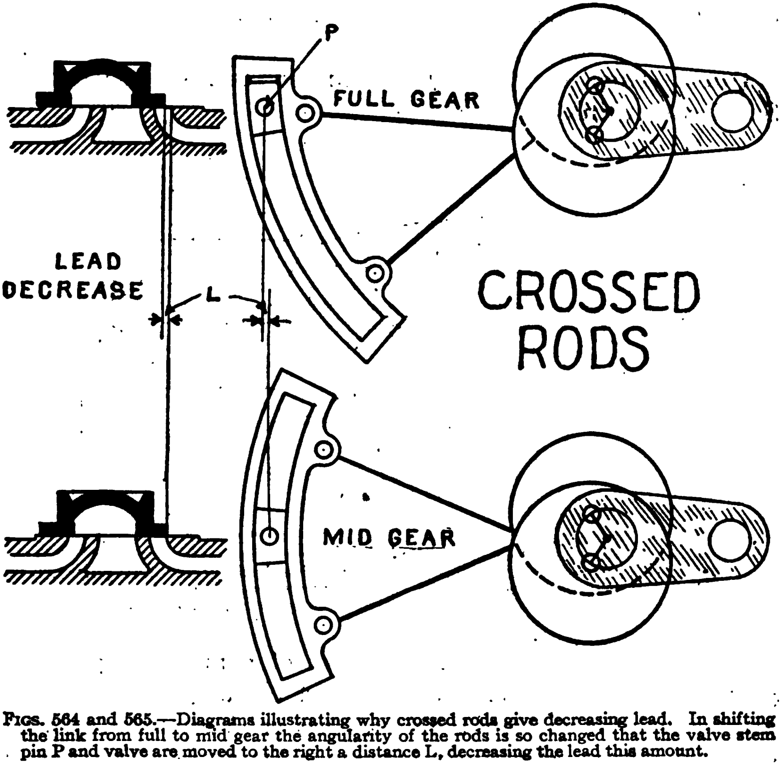
The effect of open and crossed rods is shown in figs. 562 to 565. The first two figures illustrate why the lead increases with open rods when the link is moved from full to mid gear. On account of the inclination of the rods, and the position of the eccentric centers both rods tend to push the link and valve to the left at the beginning of the movement. The upper rod, after passing the horizontal position, partially counteracts the movement imparted to the link by the lower eccentric resulting in a gradually increasing lead in amount equal to L. The position of the link center for full gear (fig. 562) should be noted.



FIGS. 562 and 563.—Diagrams illustrating why open rods give increasing lead. *In shifting* the link from full to mid gear, the angularity of the rods is so changed that the valve stem pin P, and valve are moved to the left a distance L, thus increasing the lead this amount.

Figs. 564 and 565 show why crossed rods decrease the lead from full to mid gear. The combined effect of the angularity of the rods is such that in moving the link from full to mid gear the link and pin P are moved to the right a distance L, this decreasing the lead by that amount.

In both cases the valve is shown in one of the positions with zero lead, that is, in line and line position to clearly illustrate the change in lead.



FIGS. 564 and 565.—Diagrams illustrating why crossed rods give decreasing lead. In shifting the link from full to mid gear the angularity of the rods is so changed that the valve stem pin P and valve are moved to the right a distance L, decreasing the lead this amount.

Short rods are used to emphasize the effect of open and crossed rods on the lead.

Ques. When should open and crossed rods be used?

Ans. If the link motion is intended to be used as an expansion gear, on a locomotive, open rods should be used as a greater range of expansion may be obtained with less reduction of port opening than with crossed rods.* If the link is to be used only in full gear, or in connection with an independent cut off, crossed rods may be used, and the link made straight.

Ques. How early may steam be cut off with the Stephenson link?

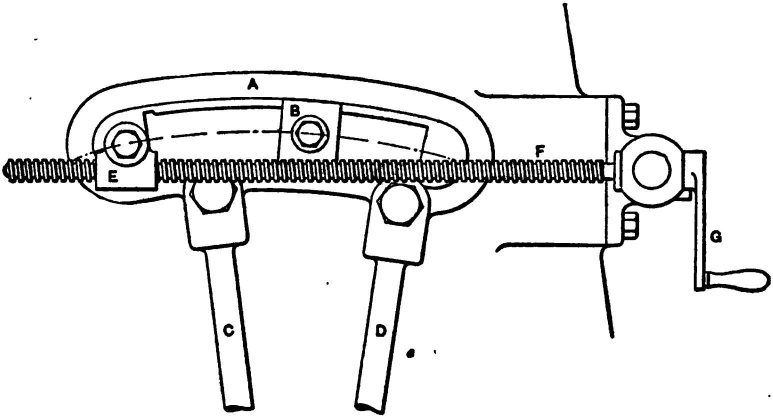


Fig. 566.—Reeves link motion adjustable cut off for engine driving centrifugal pumps, fans, etc. The cut off may be varied while the engine is in motion. The parts are: A, link; B, link block; C, D, eccentric rods; E, adjusting block; F, adjusting screw; G, adjusting crank.

Ans. This depends on the amount of port opening at full gear. If, as with locomotives, the port opening at full gear be greater than the width of the port, fairly good admission may be obtained, cutting off as early as one-quarter stroke. For shorter cut off, the admission is poor and one-sixth stroke may be taken as the minimum cut off with the ordinary valve.

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*NOTE.—On locomotives it is necessary to give little or no lead, and make the port opening greater than the port for full gear in order to prevent excessive lead and too little port opening at early cut off.

Ques. What are the different forms of the shifting link?

Ans. The slotted link as already described, the open, the double bar marine type, and the box link.

The open link is similar to the ordinary link but differs in that the eccentric pins, instead of being attached to one bar, are located as shown in fig. 567. With this construction, the eccentrics must have a larger throw, since

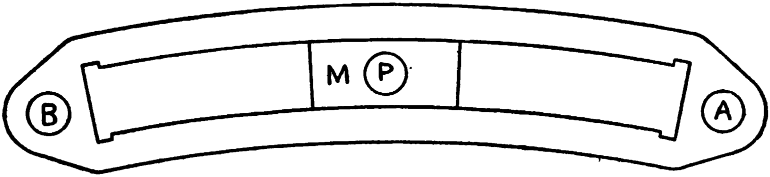
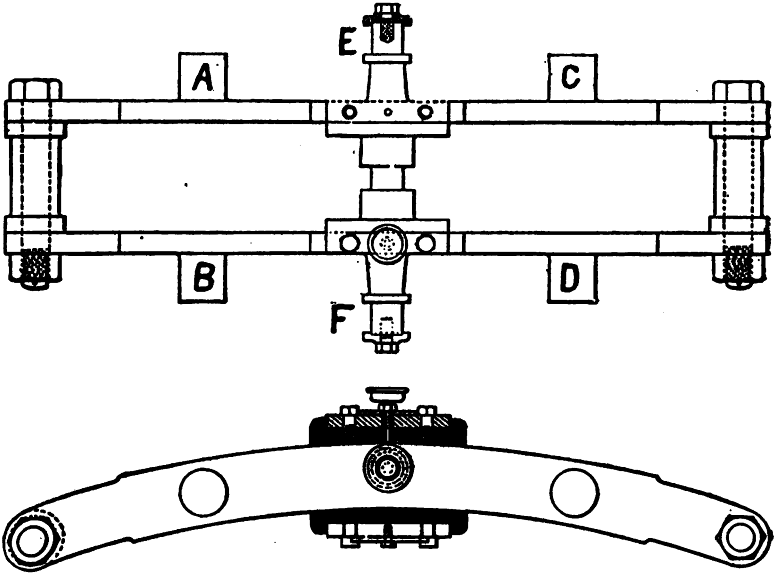


Fig. 567.—The open link; used chiefly on British locomotives where there are no rockers. The eccentric rods are pivoted at A and B, and the link suspended from the upper rod pin. The fixed point of the reach rods is below the central line of motion.



Figs. 568 and 569.—The double bar link as used on marine engines. The eccentric rods are pivoted at A, B, and C, D, on the central arc of the link which improves somewhat the steam distribution.

the eccentric pins move a greater distance than the maximum travel of the valve. The open link is used chiefly on British locomotives where there is no rocker, the link being hung from the upper eccentric rod pin with reverse shaft below the central line of motion.

The double bar marine type link is shown in figs. 568 and 569. It consists of two bars curved to the proper arc and connected at their ends by sleeve bolts which retain the bars at the desired distance apart. The eccentric rods are attached to two pairs of pins A, B, and C, D, each rod end having a double bearing. A third pair of pins E, F, receive the reach rods; these pins may be either located at the center as shown, or at the end as in fig. 575. On marine engines, an independent adjustment for cut off is frequently fitted to the high pressure cylinder valve gear, and sometimes to each cylinder.

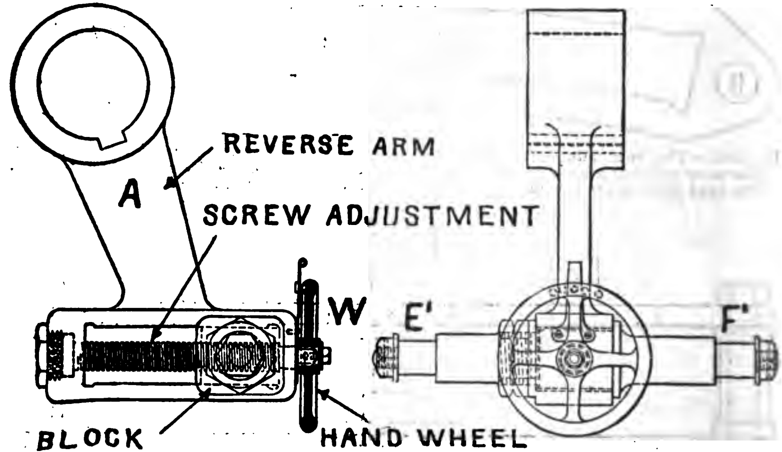


FIG. 570 and 571.—Independent cut off adjustment for link motion; usually fitted to one or more cylinders on multi-cylinder marine engines. This permits regulation of the expansions, receiver pressures, etc., so as to get a steam distribution best suited to the running conditions.

With link motion, the independent adjustment as shown in figs. 570 and 571 consists of an arm A, keyed to the reverse shaft, and having at its end, a slot within which works a block with screw adjustment.

The reach rods are attached to pins which project from the block; by turning the wheel W, the block is moved in the slot which changes the position of the link and thus alters the cut off. Fig. 572 shows the adjustable cut off arm as fitted on marine engines.

The box form of link which has the pins in the line of the slot itself is shown in fig. 573. Where a short eccentric throw is desired the box link is used to advantage. It is, however, expensive to make on account of the difficult construction.

Ques **Why is a link slot curved?**

Ans. To equalize the lead of the valve for all travels. The radius of the link is so proportioned as to make the increase or decrease of the lead the same for both strokes of the piston.*

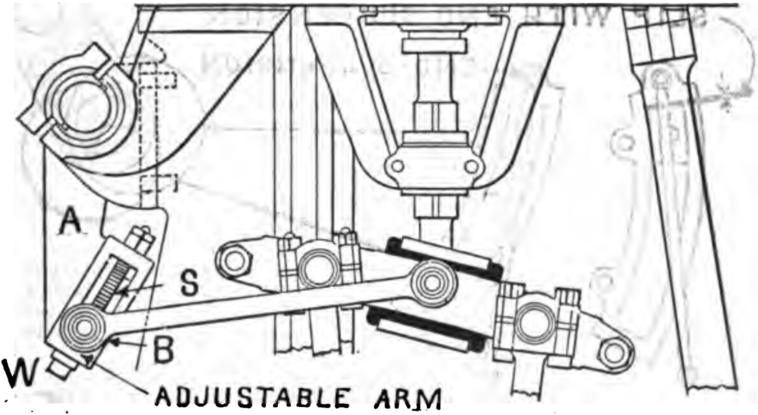
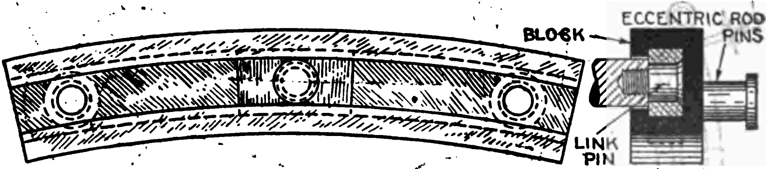


FIG. 572.—Independent cut off adjustment for link motion; view showing gear assembled on engine. The reach rods are pivoted to a block which works on the screw S. By turning this screw at W, the block is moved in the slot and the link shifted, thus changing the cut off.



FIGS. 573 and 574.—The box link. Used to advantage where a short eccentric throw is desired since the valve travel is about the same as the throw. The box link is difficult to construct.

Ques. What effect has the action of the link on release and compression?

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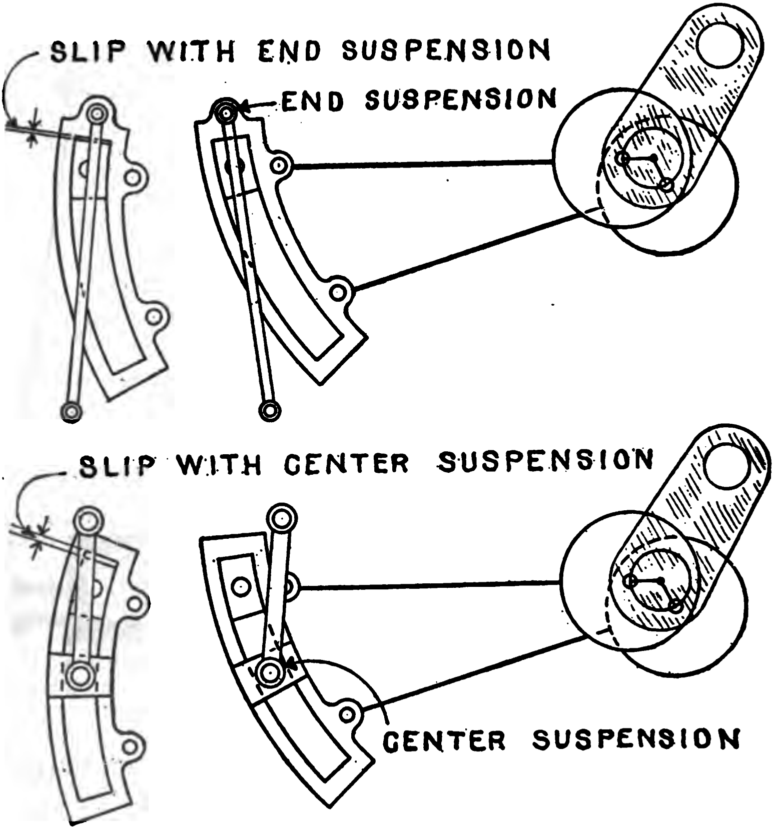
*NOTE.—The radius of the link is a little less than the length of the eccentric rod, or about twice the (lap+lead) less than the distance from the center of the link block to the center of the eccentric.

Ans. As the cut off is shortened by shifting the position of the link, these events occur earlier.

On locomotives, this peculiarity of the link motion is, within limits, an advantage, because when a locomotive is running fast, steam is cut off short, and early release and compression is desirable since owing to the



FIGS. 575 and 578.—Diagrams illustrating the effect of end, and center suspension. When possible, the point of suspension should be at the end of the link as shown in fig. 575, because the slip is less than with center suspension as in fig. 578.

high piston speed, more time is needed for pre-release, and the increased cushioning due to the early compression is absorbed in bringing the reciprocating parts to rest. The clearance space is thus filled with steam at a higher pressure, which reduces the amount of live steam required to increase the pressure in the clearance space to that of admission.

Ques. What is the "slip?"

Ans. The sliding of the link on the block which occurs during each stroke.

Ques. Why does the link slip?

Ans. The center of the block, being pivoted to the valve stem, moves in a straight line, while the ends of the reach rods which guide the link have a circular movement, hence a sidewise motion is given to the link, causing it to slip or slide on the block.

In addition to this, slip is occasioned by what might be called "the angularity of the link," that is, the inclined positions which it takes, cause a sliding action as indicated in figs. 575 to 578. It should be noted that the end of the link is furtherest from the block when the link is in the nearly vertical positions shown at the left of the figures.

Ques. What is the point of suspension?

Ans. That point where the reach rods are pivoted to the link.

Ques. What is the fixed point?

Ans. The center on the rocker arm at which the reach rods are pivoted, and about which the rods swing; the *swing center* of the reach rods.

Ques. What determines the position of the point of suspension?

Ans. The type of engine, and conditions of operation.

On locomotives the link is usually suspended near the center, but where conditions permit, the point of suspension is best located near the end.

Figs. 575 to 578 show the two locations of the point of suspension. In each figure the link is shown in two corresponding positions from which is seen the effect which changing the point of suspension has on the slip. As shown in the figure the slip is less when the link is suspended at the end, than when suspended at the center. The point of suspension is sometimes offset from the link arc, the object being to secure the minimum slip for the gear position in which the engine is mostly run.

Ques. When is the slip greatest and least?

Ans. Greatest in full gear, and least in mid gear.

Ques. What conditions tend to reduce the slip?

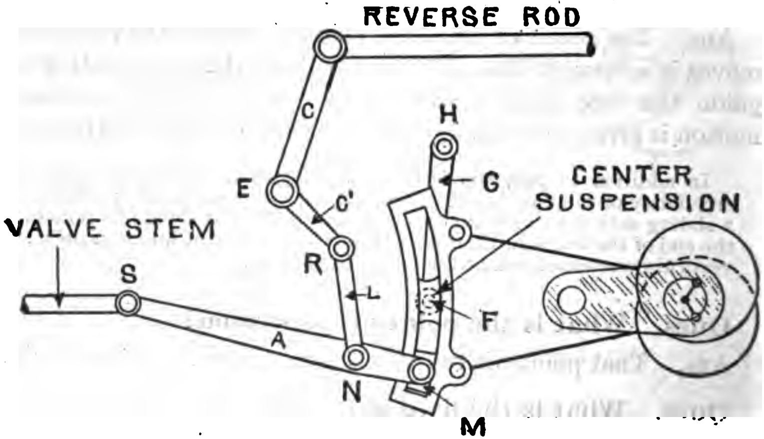


FIG. 579.—The Gooch stationary link; used chiefly where the valve requires no rocker, as on British locomotives. *The lead is constant for all cut offs.* An objection to the Gooch link is that it requires considerable distance between the shaft and cylinder on account of the long radius rod A.

Ans. Considerable angular advance, short travel, short eccentric rods, and a long link.

Ques. How long should the link be made?

Ans. It ought to be of such length that its movement in reversing is from 2½ to 3 times the travel of the valve.

The Stationary Link. Shortly after the appearance of the shifting link came the stationary link, also known as the Gooch link, named after its inventor Daniel Gooch. It has been used extensively on locomotives throughout Great Britain, and the continent, but is little used on American locomotives as it is not adapted to engines having steam chests on top the cylinders; it is especially suited to engines having no rockers.

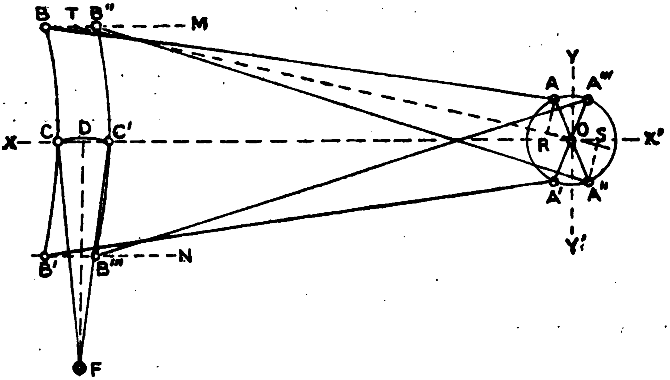


FIG. 580.—Stationary link motion. Diagram for setting the eccentrics as given by Clark. The eccentrics must be so placed as to yield the necessary linear advance of the valve, or the double of it between the positions of the link at the two ends of the stroke. Let X X', above be the center line, and O, the center of the driving axle. Through O, draw the vertical Y Y', and describe the circle A, 4½ ins. in diameter, for the path of the eccentrics. Draw the parallels M, N, 12 ins. apart, equally distant from the center line,—the centers of the link being 12 ins. apart. On the center O, with the length of the eccentric rod as radius, which in this case is assumed for convenience at 27 ins., or six times the throw of the eccentrics, cut the line M, at T, and draw TO, set off O R and O S, each equal to the linear advance of the valve,—1 5/16 ins.—and draw the perpendiculars RA, SA'', to meet the circle. Draw the diameter AA'', then OA and OA'', are the positions of the fore eccentric for the lead of the in and out strokes respectively. From A and A'' as centers, with the length of the rod, cut the line M, at B and B'', and join AB and A'' B''. These are the positions of the fore eccentric rod for the out and in strokes; and the space BB'', equal to RS, measures twice the linear advance of the valve. This construction is empirical, but it is in ordinary cases satisfactory, and the points are easily adjusted, if the interval BB'', be not exactly equal to twice the linear advance. The position of the back eccentric at A' and A''', is found by drawing parallels to the vertical Y Y', through the points A and A''. The lower centers of the link, at B' and B''', are found similarly to the centers at B and B''. Draw BCB' and B'', C' B''', for the relative positions of the link. From C and C', as centers, with the length of the sustaining link as radius, find the point of intersection F, the position of the fulcrum, over which the link will vibrate equally on both sides of the vertical FD. The linear advance of the eccentrics, that is, the perpendicular distance of their revolving centers from the vertical CD, does not exceed seven-eighths inch, which is nevertheless sufficient, aided by the obliquity of the rods, to cause an advance of 1 5/16 ins. at the link. Applying the same method to find the set of the eccentrics for the 54 inch rods of the valve motion already illustrated, the advance of the eccentrics is exactly 1.075 inches, or over 1 1/16 ins., for 1 5/16 ins. of advance of valve. The open forms of link require a like process for the setting of the eccentrics.

The stationary link requires considerable distance between the shaft and valve by reason of the long radius rod necessary between the link and valve stem. Its feature with respect to the steam distribution is that it gives *constant lead for all cut offs*, with either open or crossed rods.

The concave side of the link is turned toward the valve as shown in fig. 579 the radius of the link being equal to the length of the radius rod A. To reverse the engine, the block M, is moved in the slot by the lever C, and reverse arm C', both keyed to the reverse shaft E, the movement being transmitted to the radius rod A, through the reverse link L. The link is suspended at F, by the reach rod G, which is pivoted at the fixed point H. Since the radius of the link is equal to the length of the radius rod A, it is evident that the block may be shifted from one end of the slot to the other without moving the point S, therefore the lead remains constant for all degrees of expansion.

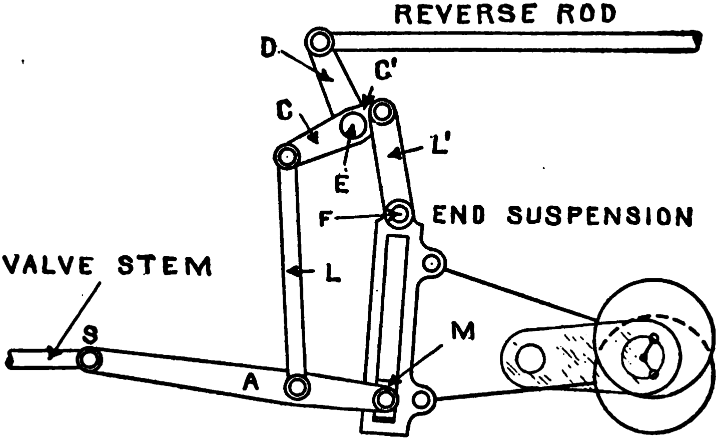


FIG. 581.—The Allen straight slot link; a modification of the Gooch link, and designed to secure equal steam distribution at each end of the cylinder.

The Allen Link.—This form of link motion invented by Alexander Allen was designed to combine the leading features of both the Stephenson, and Gooch links. It is so constructed that the parts are almost balanced, hence on locomotives it does not require equalizing springs, or counterweights.

As shown in fig. 581, it consists of a straight link, with a radius rod A, and block M; both link and rod are moved by a double suspension lever, or rocker with arms C, C′, attached to the reverse shaft E. Since the link is straight, the center of travel of the block varies, but this is compensated for by the effect of changing the slant of the radius rod A.

The position of the link and radius rod is shifted by means of a third arm D, attached to the reverse shaft E. The proper proportioning of the two arms C, C′ is an important point in the design of this link motion.

Ques. What effect has the Allen link motion on the lead?

Ans. With crossed rods, the lead decreases as the cut off is shortened.

Ques. Is the variation of lead greater with the Allen or Stephenson gear?

Ans. With the Stephenson gear; a well proportioned Allen gear, having a long radius rod and short travel, will give practically constant lead.

Ques. What are the advantages of the Allen link?

Ans. The parts being in balance require no equalizing springs or counterweights; the slip is small.

Ques. What disadvantages does the Allen gear possess?

Ans. It requires considerable distance between the valve and the shaft on account of the radius rod. The lead is constant, which is not desirable for locomotives. More parts are required than with other types of link.

The Allen link is specially adapted for use on inside connected locomotives, that is, locomotives having steam chests at the side of the cylinders although a modified form of the Allen gear has been used on American locomotives.

The Fink Link.—This is a simple form of link motion and is used on the Porter-Allen engine. The lead is constant, and its principles of operation are illustrated in fig. 582.

The link forms a part of the eccentric strap, and is suspended at F, the fixed point being below at B. Cut off is varied by shifting the position of the block M, to which is pivoted the steam radius rod D. The figure shows a separate exhaust radius rod D′, which is set permanently in full gear; this rod operates the exhaust valves independently of the steam valves.

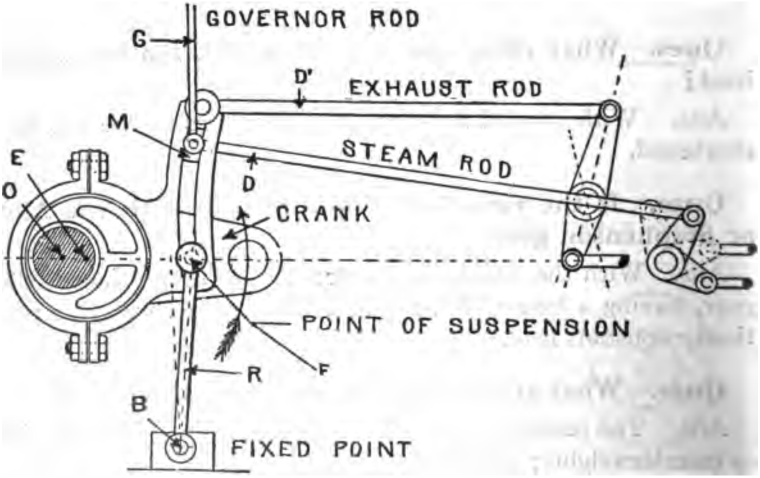


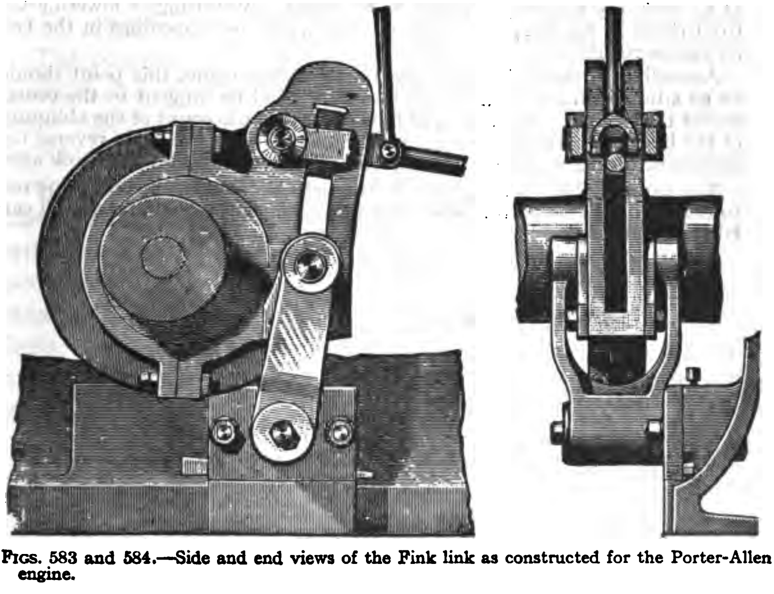
Fig. 582.—The Fink link; a simple form of motion. Its special feature is the long range cut off, obtained without disturbing the motion of the exhaust valve.

Cut off is made automatic by a connection G, to the governor. When the Fink link is used as a reversing gear, the point of suspension F, is placed at the intersection of the line of centers, and a perpendicular to it through the center of the link block when in full gear.

The link receives a peculiar motion on account of the horizontal and the vertical throws of the eccentric. The horizontal throw alone only moves it from one to the other of the lead lines, which motion only draws off the lap of the valves.

The opening movement is produced by the tipping of the link alternately in the opposite directions beyond the lead lines, these tipping motions being given by the vertical throws of the eccentric.

The upward throw tips the link in the direction from the shaft, and opens the port at the further end of the cylinder; and the downward throw tips the link towards the shaft, and opens the port at the crank end of the cylinder. At the same time its horizontal, throw is drawing the valve back, and when, in this return movement, that point in the link at which the block stands, crosses the lead line, steam is cut off.



Figs. 583 and 584.—Side and end views of the Fink link as constructed for the Porter-Allen engine.

Figs. 583 and 584 show a side and end elevation of the Fink link as designed for the Porter-Allen engine. It should be noted that the link is suspended from both sides, thus avoiding any lateral stresses. The range of cut off is from zero to six-tenths of the stroke. The link is especially suited to a long range cut off since the exhaust features are not affected by the degree of admission.

The exhaust valves open and close their ports in such a manner that the opening is made while the valve is moving swiftly, and one-half of the opening movement has been accomplished when the piston arrives at the end of its stroke.

The valves are so constructed that this portion of the movement opens the full area of the port, which does not begin to be contracted again until the center line of the link has recrossed the lead lines on its return. The speed of the piston is then also diminishing, and the exhaust is not throttled at all until the port is just about to be closed. By raising or lowering the fixed point B, fig. 582, the equality of lead, and port openings in the two strokes is regulated.

According to the builders of the Porter-Allen engine, this point should be so adjusted that the arc of motion at F, shall be tangent to the center line of the engine. The motion of F, is distorted on account of the obliquity of the line F E. To neutralize this, the makers use a rocker to reverse the motion of the valve, and put the center of the eccentric on the crank axis.

The ratio of F E, to O E, is made the same as the ratio of connecting rod to crank so that one error offsets the other, hence the lead and cut off can both be equalized.

CHAPTER 10

REVERSING VALVE GEARS; RADIAL VALVE MOTIONS

The object sought in the introduction of the so called radial gears is to overcome the defects of the shifting link gear, and in some cases to obtain a more accessible gear.

While the shifting link gives equal lead for various cut offs it does not give constant lead, that is, as the cut off is shortened, the lead increases or diminishes according as the link is arranged with open or crossed rods. Moreover, premature compression occurs as the cut off is shortened. These distortions in the steam distribution are in many cases undesirable. These defects, and the desire to avoid a multiplicity of eccentrics on multi-cylinder engines, are the chief reasons for the adoption of radial gears.

The better steam distribution secured by radial gears is in some cases more or less offset by complicated construction consisting of numerous parts and joints subject to wear.*

General Principle of Radial Gears.—The object sought in the invention of radial gears is *to obtain from some reciprocating or revolving piece of the engine, an arrangement of mechanism*

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*NOTE.—According to Sothern (Principal, Sothern's Marine Engineering College, Glasgow), the general experience of engineers is that the disadvantages of radial gears more than balance the advantages, with the result that the ordinary link motion will be found fitted in even the most modern and up to date marine engines, as being simpler and more reliable.

a point in which shall describe an oval curve, and by altering the direction of the axes of this curve, to produce variable cut off and reversal.

Accordingly, a radial gear may be defined as *one in which the motion of the valve is taken from some point in a vibrating rod, one end of which moves in a* ***closed curve,*** *while a third point on the rod moves in a* ***straight line*** *or* ***open curve.***

Hackworth Gear.—This gear, which was invented by John W. Hackworth, and patented in 1859 and 1876, was the first radial gear and it probably gave rise to all modern radial gears.

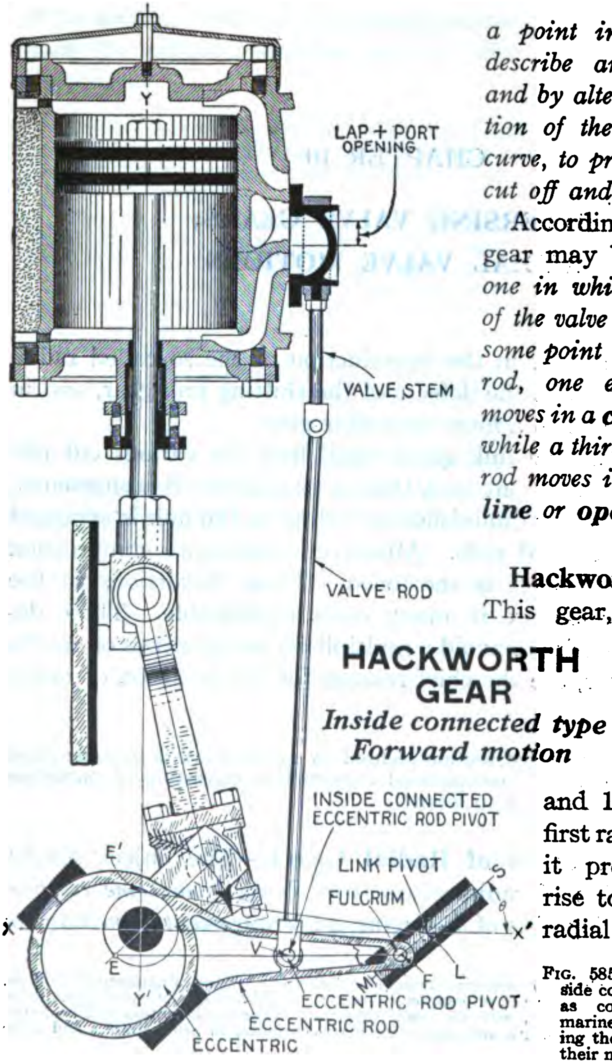


FIG. 585.—Hackworth inside connected valve gear as constructed for a marine engine; view showing the various parts and their names.

The principle of the Hackworth gear, as stated by Seaton is as follows: *"The motion of a point on a rod, one end of which moves in a* **circle,** *and the other on a* **straight line** *passing through the center of that circle, is on an* **ellipse** *whose major axis coincides with the straight line. If, however, the end of the rod slide on a line inclined to this center line, the major axis of the ellipse will be inclined."*

There are two types of Hackworth gear which may be classified as:

1. Inside connected;
2. Outside connected;

according as the valve rod is connected to the eccentric rod between the eccentric and eccentric rod pivot, or beyond the eccentric rod pivot.

Fig. 585 shows the inside connected type as arranged for outside admission. Its essential parts are: eccentric, eccentric rod, link, valve rod, eccentric rod pivot, link pivot, and valve rod pivot, and means for shifting and securing the link in any position within its arc of adjustment. As shown, the center of the eccentric is at E, or opposite the crank, corresponding to 90 degrees angular advance.

The link consists of a straight slot and guides a reciprocating block which is pivoted to the end of the eccentric rod at F. The pivot L, of the link is located in the line X X′; which passes through the center of the shaft perpendicular to the cylinder axis Y Y′. The point of cut off and direction of rotation of the engine depend upon the angular position of the link with respect to the axis X X′.

The location of pivot L, and length E B, of the eccentric rod is such that E B = E′ L. When these two distances are equal, F, will coincide with the center L, of the link when the connecting rod is on either dead center, and the slotted link may be turned from full gear forward through its horizontal position to full gear reverse without moving the valve. Hence, when the lap is the same on both ends of the valve the leads are constant for all positions of the link, and consequently for all cut off.

The valve is set by adjusting the valve stem for equal lead.

The correct location of the valve rod pivot V, is necessary to secure proper steam distribution. V, must be so located that when the engine is on the dead center and the link is in its horizontal position the distance from V to the horizontal axis X X′ = *lap* + ***lead***, as shown in fig. 586.

The operation of the gear may perhaps be better presented graphically than by description. Thus, figs. 587 to 590 show the positions of the gear in full forward motion for the principal events of the stroke, viz.: lead, cut off, pre-release, and compression, and figs. 591 to 594 similar positions in full reverse motion.

Figs. 595 to 597 show the gear in full forward motion for various inclinations of the link giving various cuts off.

In addition to permitting reversal with only one eccentric, the chief advantages of this gear are its quick motion at the point of cut off and the large range of cut off possible, wire drawing from small opening and slow closing of the port, as is the case with the shifting link motion.

The chief objections to Hackworth's gear are:

1. The friction and wear of the block and link especially when the link is inclined to the XX′ axis;

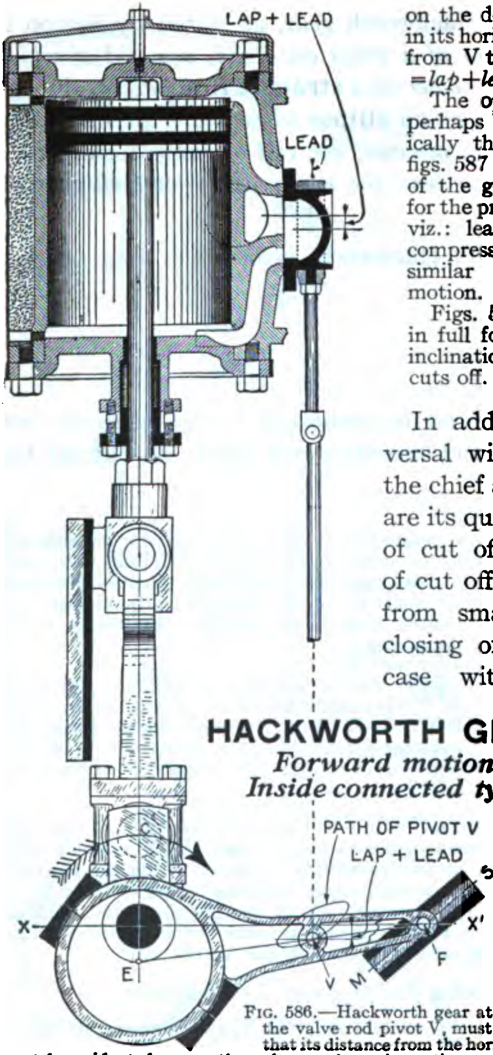
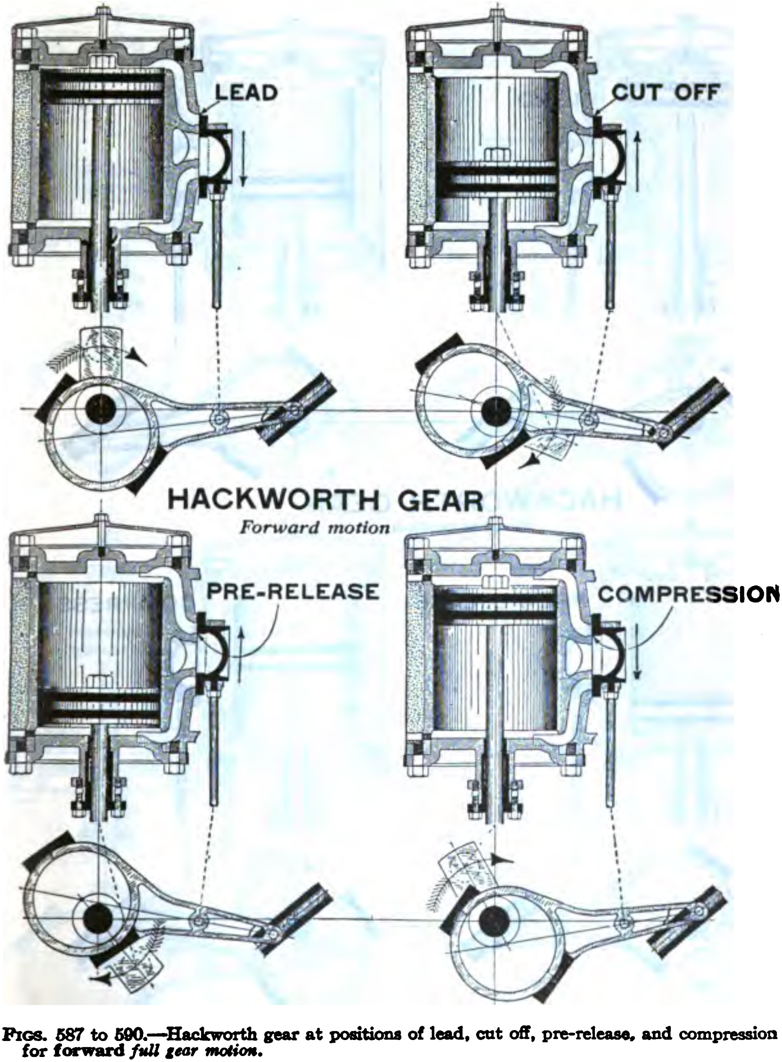
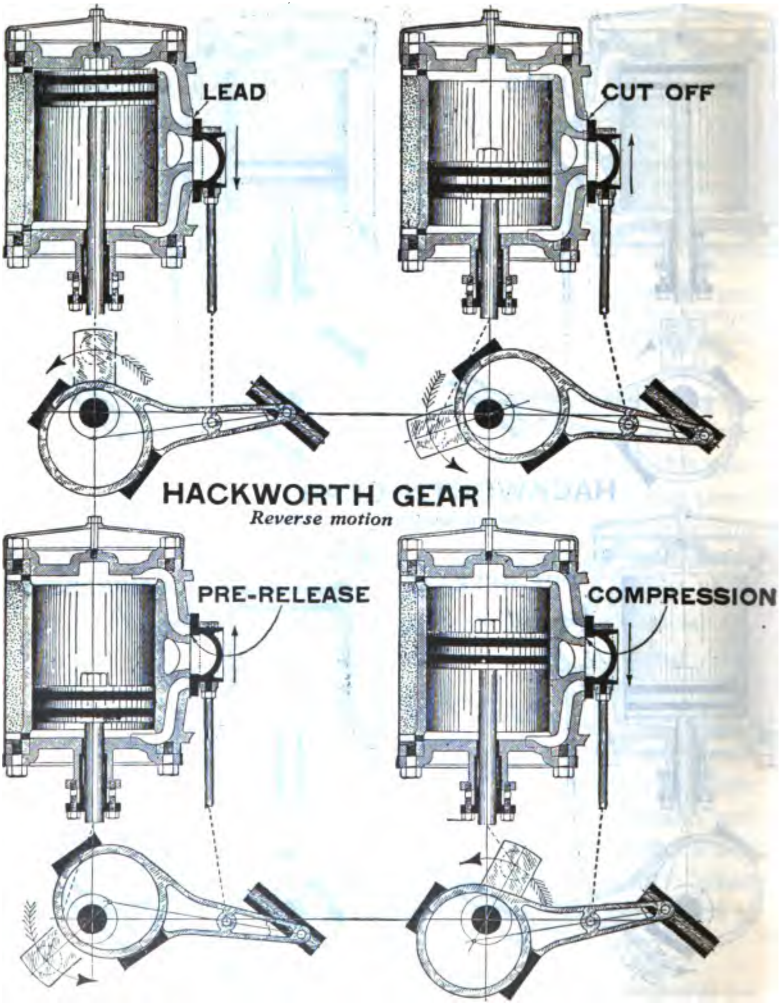


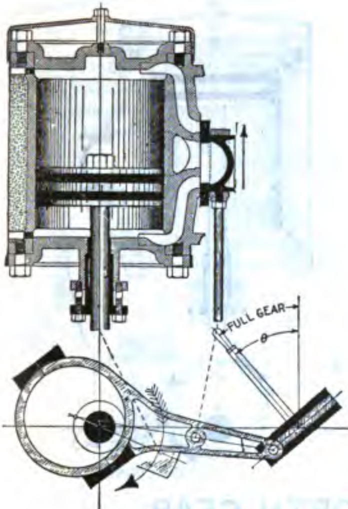
FIG. 586.—Hackworth gear at dead center position showing that the valve rod pivot V, must be so located on the eccentric rod that its distance from the horizontal axis X X′ = *lap* + *lead*. This must be evident, because the valve as shown in section must have its proper *linear advance*.

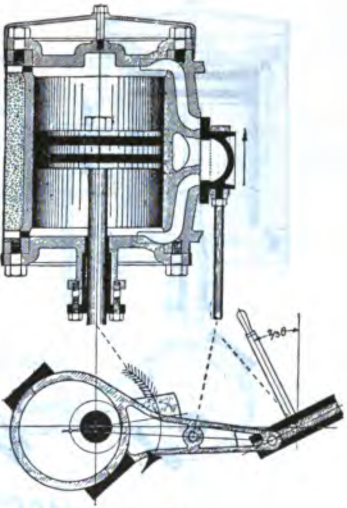


Figs. 587 to 590.—Hackworth gear at positions of lead, cut off, pre-release, and compression for forward ***full gear motion.***



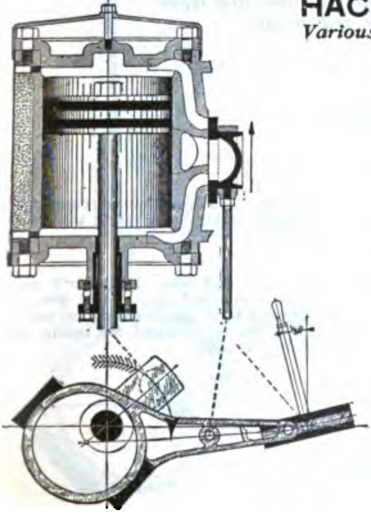
Figs. 591 to 594.—Hackworth gear at positions of lead, cut off; pre-release, and compression, for **reverse** *full gear motion.*



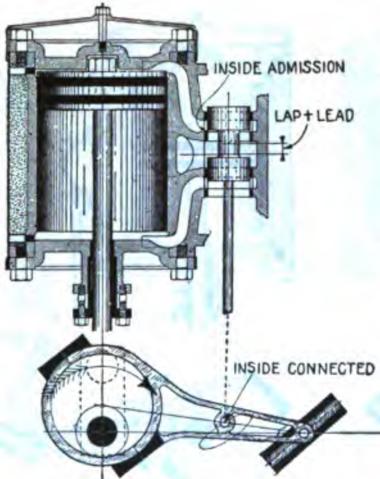


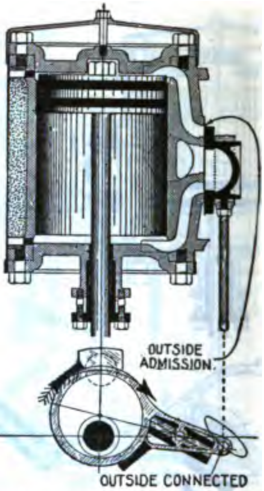
HACKWORTH GEAR

Various cut offs; forward motion



FIGS. 595 to 597.—Hackworth gear at various cut off positions **forward** *motion.*





HACKWORTH GEAR
Inside and outside connected types
Forward motion

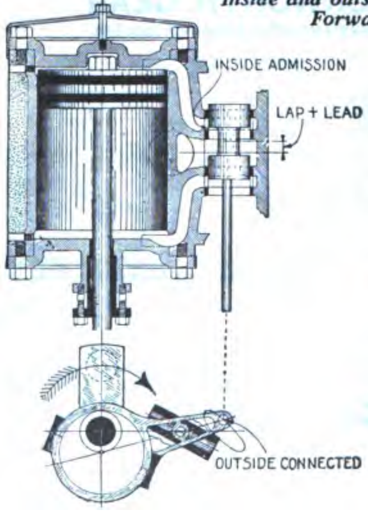


Fig. 598.—Hackworth inside connected gear as arranged for inside admission valve.

Figs. 599 and 600.—Hackworth outside connected gear; fig. 599, as arranged for outside admission valve; fig. 600, as arranged for inside admission valve.

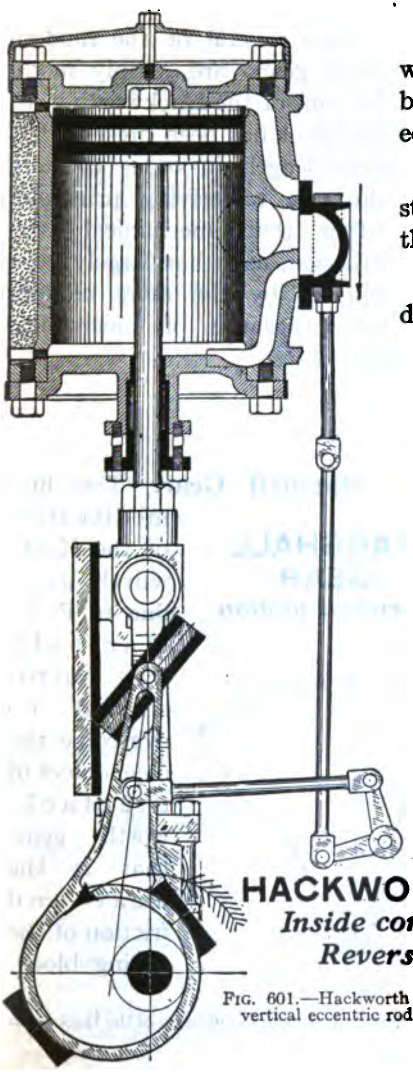
2. Large eccentric necessary when valve rod pivot is located between the eccentric and eccentric rod pivot;

3. Considerable traverse stress on the eccentric when the valve is unbalanced;

4. Numerous pins, liable to derangement.

In later designs the first objection was overcome by using rollers instead of a sliding block. The gear has worked fairly well, and for engines of small power, has been found a convenient arrangement, especially when much variation in cut off is required.

Fig. 598 shows the inside connected gear arranged for inside admission. Here the eccentric is in line with the crank, 180° from its position for outside admission, that is, its angular advance is made—90° instead of +90°. Figs. 599 and 600 show the outside connected gear arranged respectively for outside and inside admission.



HACKWORTH GEAR
Inside connected type
Reverse motion

FIG. 601.—Hackworth inside connected gear as arranged with vertical eccentric rod.

Since several of the modern radial gears are simply modifications of the Hackworth gear, the latter has been presented at some length in order to fully illustrate underlying principles rather than mechanical construction, as it has largely been replaced by the more modern forms because of mechanical difficulties.

Marshall Gear.—This first modification of the Hackworth gear, due to F. C. Marshall, was introduced to overcome the chief defect of the Hackworth gear, that is the wear and friction of the sliding block.

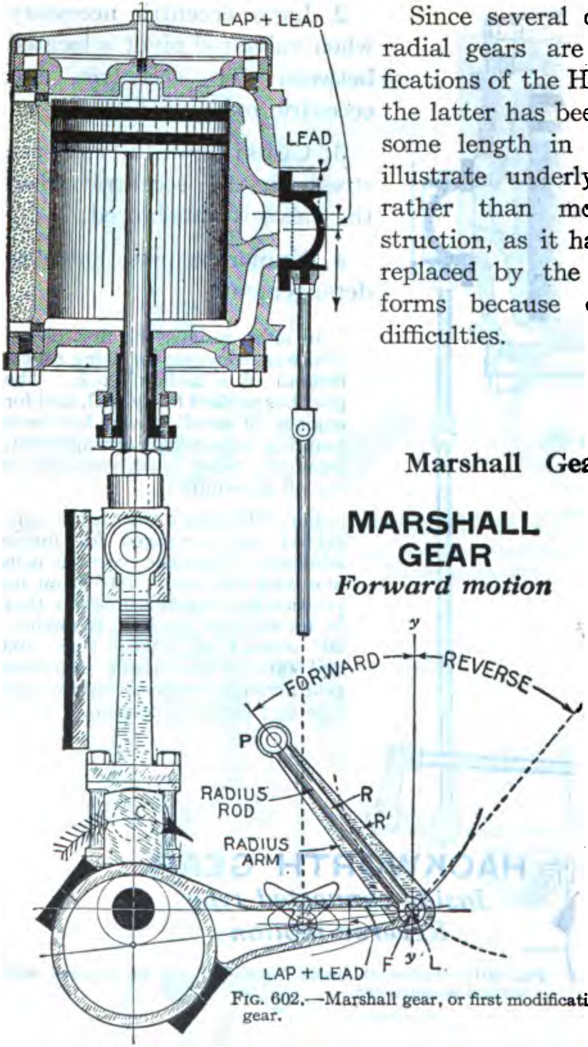


FIG. 602.—Marshall gear, or first modification of the Hackworth gear.

It is simply the Hackworth gear *with a swinging arm substituted for the link* as shown in figs. 602 and 603, the other parts are exactly as Hackworth arranged them.

Here the eccentric rod pivot F, located *at the* **end** *of the rod* is attached to a suspension or radius rod R, the other end of which is pivoted at P, to a radius arm R′, which turns about L, and whose angular position with respect to the central axis *y y′*, controls the point of cut off and direction or rotation. In construction, a geared quadrant is attached to the radius arm to provide means for setting the radius arm in any position within its arc of adjustment.

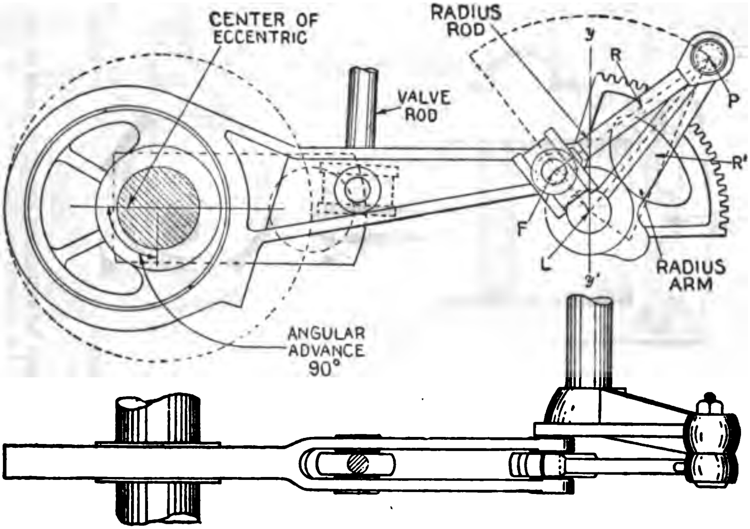


Fig. 603.—Construction detail of Marshall valve gear showing general proportion of parts.

The pivot L, is located precisely as in the Hackworth gear.

Since P F, is made equal to P L, the arc described by F, will pass through L, for all angular positions of the radius arm, and F, will coincide with L, when the engine is on either dead center. Hence, if the laps be the same the lead will be constant and equal.

The motion of the valve is the resultant of the two vertical components of motion due to the eccentric and radius arms acting at the ends of the eccentric rod.

The steam distribution of the Marshall gear is not so good as with the

Hackworth gear, due to unequal travel of the valve on each side of mid position; this necessitates double ports at the crank end of the slide valve in order to get as much port opening as at the head end.

The eccentricity of eccentric should be small as possible to reduce inequality of distribution at the two ends. Cut off cannot be equalized for all degrees of expansion, but can be adjusted for some given cut off and the others will be approximately correct. A slight angular advance may be given the eccentric with advantage, sacrificing at the same time equality of lead for different cut offs. This

MARSHALL GEAR
Forward and reverse motions

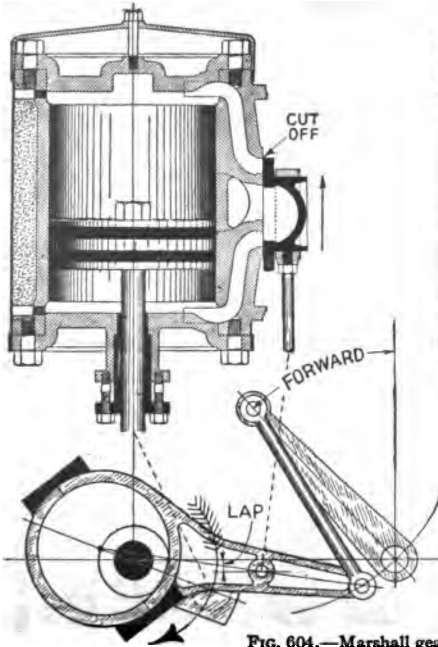


FIG. 604.—Marshall gear in cut off position for ***latest cut off; forward gear.***

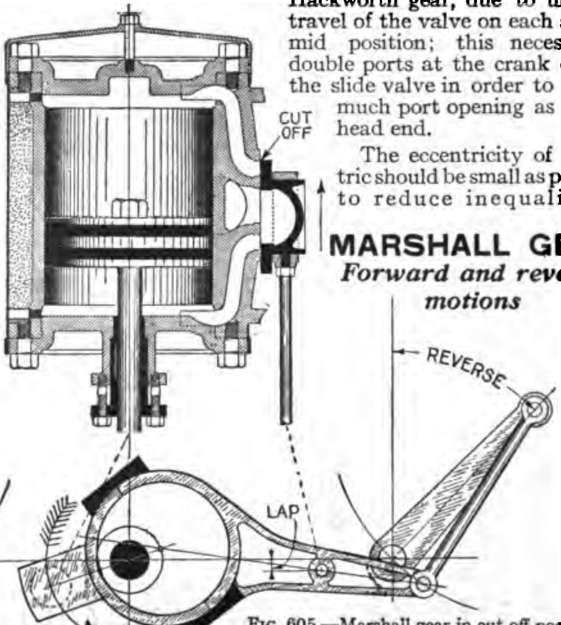


FIG. 605.—Marshall gear in cut off position for ***latest cut off; reverse gear.***

will be found to improve the distribution very much otherwise. The direction of this angular advance will depend on the position of the reversing quadrant and the connections of the eccentric lever, and must be determined for the special case.

The Marshall gear is the simplest of the radial valve gears; its chief recommendation lies in the small number of working parts and the excellent distribution of steam but the so called Stephenson link may be designed to accomplish almost as much. More opening must be given to the crank end to get equal steam distribution.

MARSHALL GEAR

Port openings forward and reverse

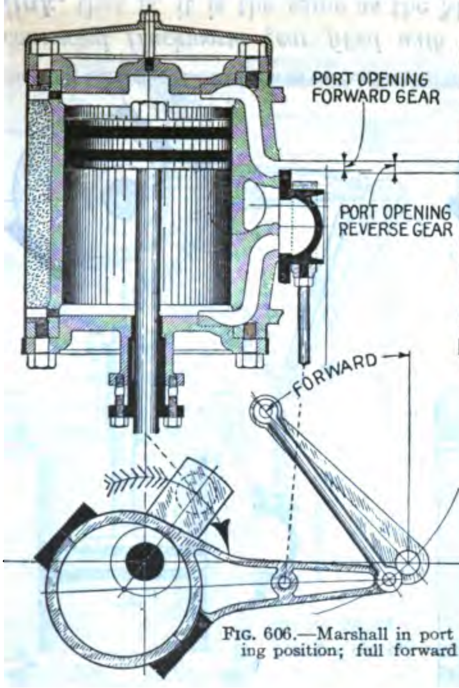


FIG. 606.—Marshall in port opening position; full forward gear.

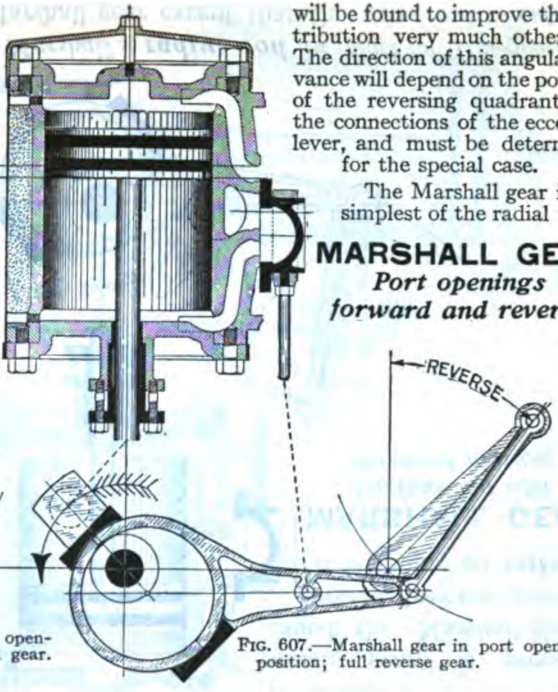
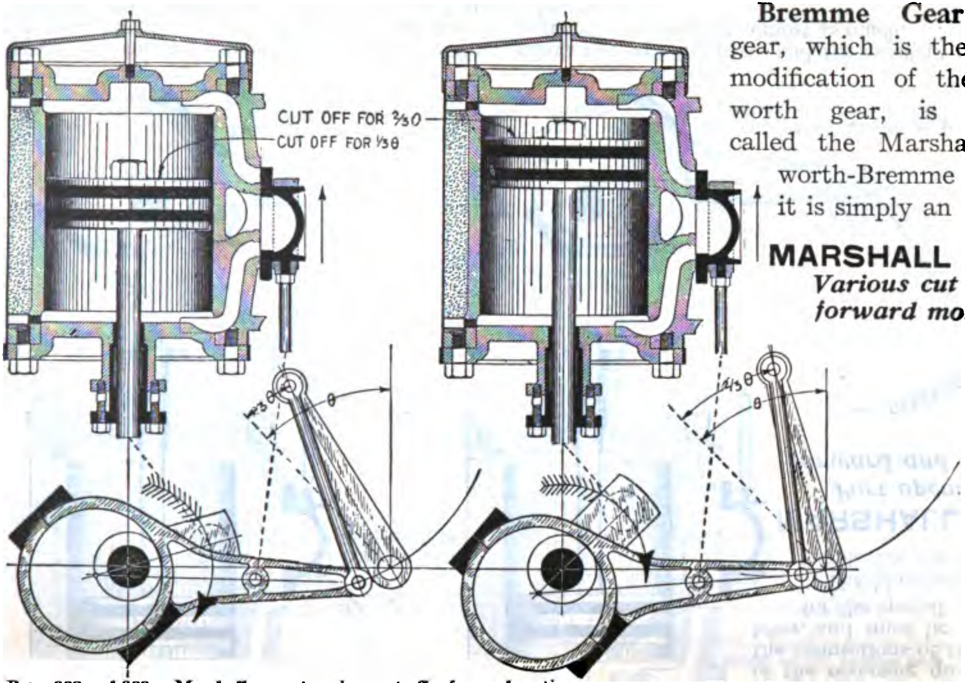


FIG. 607.—Marshall gear in port opening position; full reverse gear.



FIGS. 608 and 609.—Marshall gear at various cut offs; forward motion.

Bremme Gear.—This gear, which is the second modification of the Hackworth gear, is properly called the Marshall-Hackworth-Bremme gear as it is simply an ***outside connected Hackworth gear fitted with Marshall's*** ***radius rod*** *in place of Hackworth's* ***link,*** that is, it is the same as the Marshall gear except that the valve rod is *outside* *connected*, or pivoted to the end instead of the middle of the eccentric rod.

The gear, as constructed for a marine engine is shown in fig. 610, and consists of a single eccentric E, which either has to be set directly opposite the crank C, or in the same direction with the crank, according to the design of the valve gear.

The eccentric operates the eccentric rod L, which also forms one-half of the eccentric strap J, the extreme end of this lever is attached to the valve rod, by means of a pivot N, and thus to the valve stem.

The fulcrum of the eccentric rod is at F, about which it is swung vertically by the throw of the eccentric, the amount of travel thus imparted to the valve being equal to the *lap and lead* for both ports.

The travel necessary to open the port is imparted to the valve through port N, by the up and down motion of the fulcrum F, due to the horizontal throw of the eccentric, which causes the radius rod R, pivoted at K, to swing, and thus raise and lower the fulcrum.

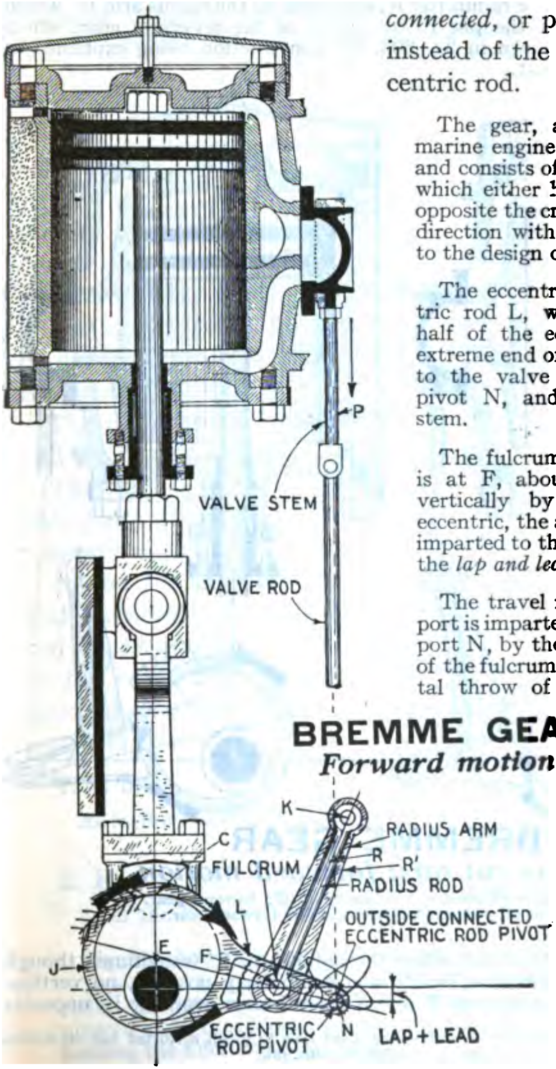
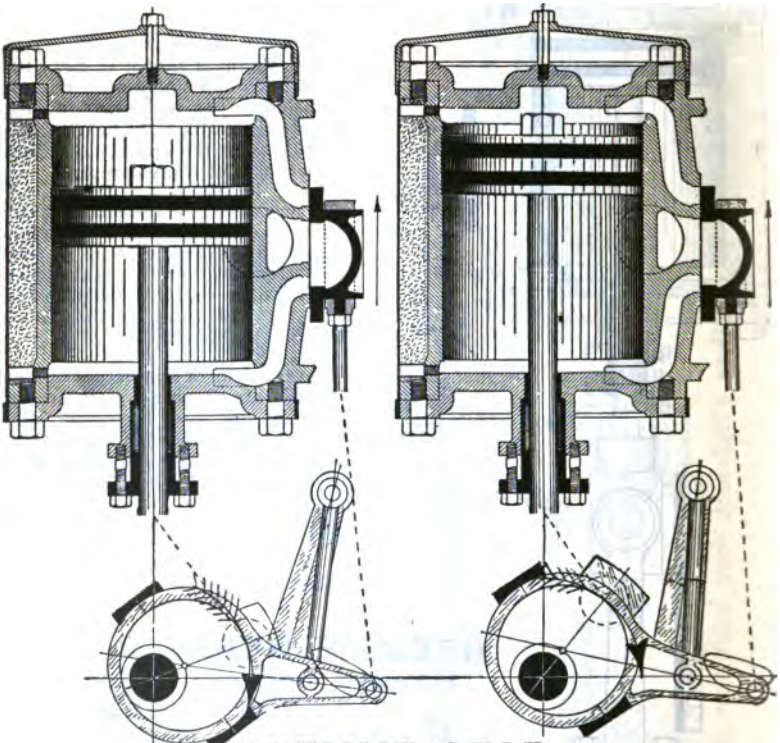


Fig. 610.—Bremme gear, or second modification of the Hackworth gear.

The upper end of the radius rod R, is pivoted to the radius arm R′, which can be swung about the pin F, by means of the reversing gear, which is similar to that shown in fig. 603, the construction being explained in the accompanying text.



BREMME GEAR

Various cut offs; forward motion

Fig. 611.—Bremme gear in cut off position for latest cut off; forward gear.
Fig. 612.—Bremme gear in cut off position for short cut off; forward gear.

It must be understood that when the radius arm R′ (sometimes, though ill-advisedly called tumbling link*) is at its midway position, no vertical motion is given to the fulcrum F, and if it be thrown over into its opposite position the motion is reversed to that indicated in the figure. By reducing the inclination of the arm, the cut off is shortened.

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*NOTE.—The author prefers to confine the word link to mean a slotted bar or device wherein a block slides as in the shifting or so called Stevens link.

According to the inventor, the angle of the reversing lever (that is, the radius arm R′, fig. 608) from the central line and known as the "deviation line" should never exceed 25° on either side.

Joy Gear.—This gear invented by David Joy in 1880 is perhaps the best known of the radial gears, and avoids altogether the use of eccentrics. Its motion is superior to that of the ordinary eccentric in that the parts are opened and closed rapidly with slow valve movement during expansion and exhaust.

The lead is constant, and the cut off nearly equalized for all grades of expansion. The compression is less at short cut off than with link motion.

The Joy gear has been extensively used on English locomotives and on marine engines.

The chief objections to the gear are its great number of parts and joints which are in the way and subject to wear. In design the various pins should be made substantial.

In the Joy gear motion is obtained from the connecting rod and imparted to one end of what corresponds to the eccentric rod in the previous gears, the other end of which is connected to the valve rod. There are two types of Joy gear, classified according as the motion received from the connecting rod is modified by:

1. A link;

   As in the Hackworth gear, or

2. Radius rod

   as in the Bremme gear.

Fig. 613 shows the link type as applied to a marine engine.

The lever A, (previously mentioned as corresponding to the eccentric rod in the radial gears already described, especially the Bremme gear), is pivoted at B, to a block arranged to slide in the curved link L, the pivot forming the fulcrum of the lever A.

Motion is imparted to the lever A, directly from the connecting rod by means of the rod C, one end of which is pivoted to the connecting rod, the other end to the rod D.

The vertical motion of the rod C, moves the valve an amount equal to its *lap* + *lead*, while the horizontal motion causes the ports to open their full opening by moving the fulcrum up and down in the inclined link.

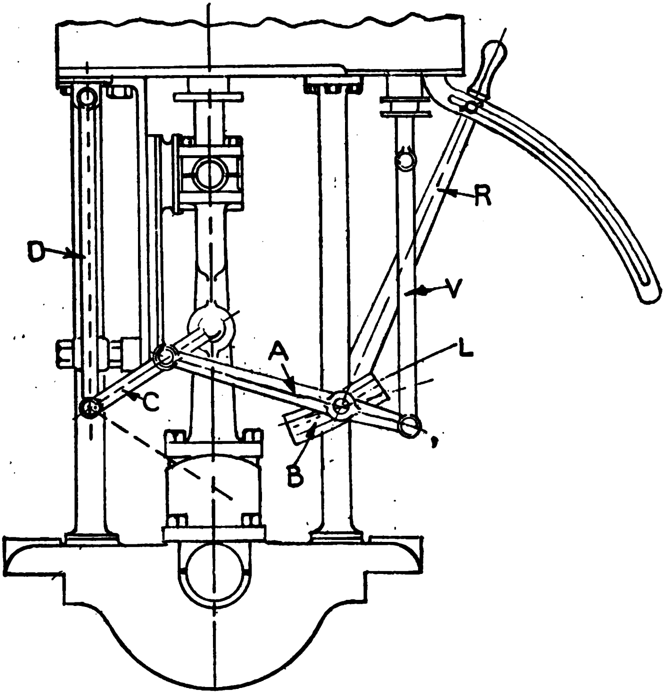

FIG. 613.—Joy valve gear, link type. In this gear, as can be seen, no eccentric is employed, the motion being taken from the connecting rod, thus permitting more liberal main bearings.

By means of the reversing lever R, the inclination of the link L, can be altered, or reversed, to vary the cut off or reverse the engine.

Fig. 614 shows the radius rod type of Joy gear as applied to a marine engine.

Diagram for Setting Out Joy's Valve Gear.—The following is the method of design as given by the inventor:

"On the connecting rod AB, fig. 615, take a point C, so that its total vertical vibration DD', is not less than twice the full valve travel, perferably a little more. Through DD', draw XX, perpendicular to AB, and at the proper distance from AB, lay down the center line of the valve spindle. Mark the extreme position of the point C, for inner and outer dead centers, and choose such a lever CE, whose total angle of vibration CEC', does not exceed 90°, and carry the end E, by an anchor link

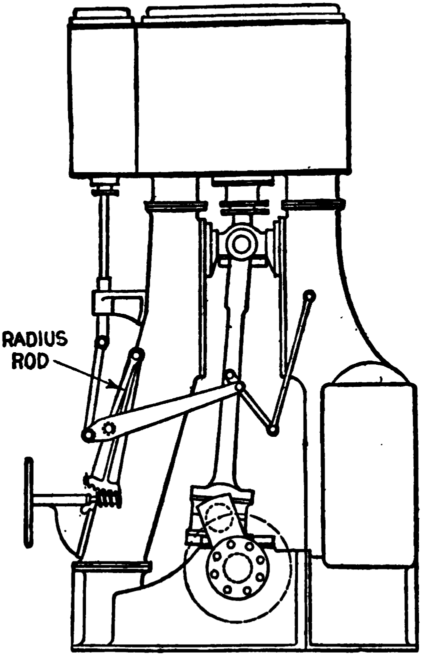

FIG. 614.—Joy valve gear, radius rod type. The substituting of a radius rod in place of the link avoids the objectional friction and wear of the latter.

E F, the mid position of which is parallel to the connecting rod when horizontal.

"Next, on the center line of the valve spindle produced, and on each side of the vertical X X, mark off the points J J', each distant from the vertical by an amount equal to lap and lead. From the point J, draw a line J H, the center line of a link that, by virtue of its connection to C E, will move the point K (the point where J H crosses the vertical) equally on each side of the central point K. The point K, is the center of oscillation of the curved guides in which slide the blocks carrying the fulcrum M, of the lever J H. The position of H, is best found by a tentative process, and to test whether the chosen point be a correct one, the equal vibration required is marked off on each side of K, and lettered L L. The distance LL, is equal to the vertical vibration of the point C, on the connecting rod, that is D D'. From D, as center, and with C H, as radius, mark the point M, on the vertical X X, and with the same radius, mark off N, from D', M and N, are the positions of H, when the engine is at

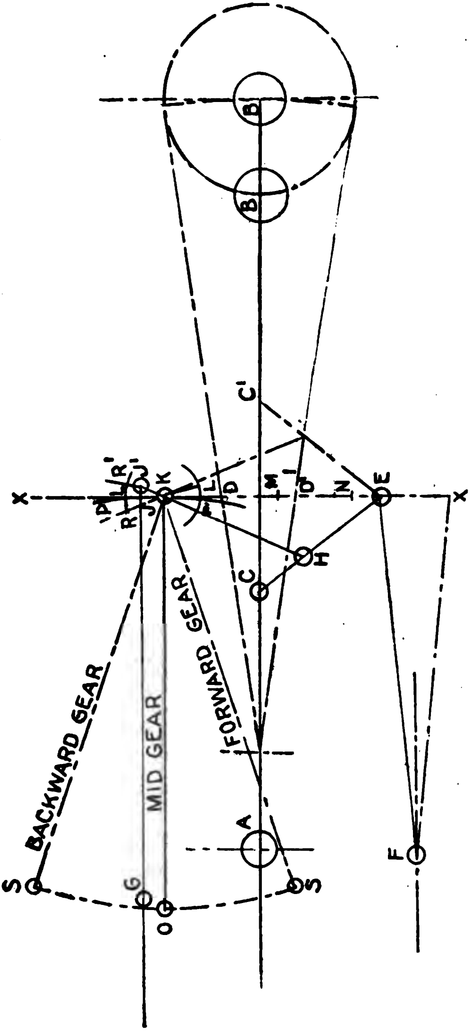

FIG. 615.—Diagram for laying out the Joy valve gear according to directions of the inventor as quoted in the accompanying test.

half stroke, or thereabouts, and these points give the total vertical vibration of H.

"From M, as center, and with HK, as radius, describe an arc cutting the vertical XX, and from N, as center and with the same radius describe an arc also cutting the vertical XX, then, if the point H, be the correct one, the arcs just drawn cut the vertical XX, at L and L. Should the points of intersection fall *below* L and L, H is too near E, but if they fall above L and L, H is too near C. The exact position of H, is generally found by a second trial. The valve rod JG, may be of any convenient length, but the center line of the slides must be struck with the same radius. From the point K, draw a line KO, parallel to AB, and with center on this line, and with JG, as radius, describe an arc containing K, and cutting the curves LL, struck from K, as center, in PP′.

"From P, or P′, and on each side thereof, mark off on the arcs LL, an amount equal to 1¼ times the maximum port opening required, and let RR′, be the points. With centers on the arc SS, struck from center K, describe arcs passing through K R and K R′, these arcs represent the center lines of the curved slots for forward and backward gear, and when the latter are in either of these positions the point of cut off is about 75%. Should a later cut off be required the slots must be carried still further from their vertical position.

"It is seen in the diagram that the fulcrum K, of the lever JH, coincides with the center of oscillation of the curved slots or guides when the crank is on either dead center. Evidently when these points coincide, the angle of the guide can be altered to any extent without disturbing any other part of the mechanism, a state of things which shows that the lead is constant."

The inventor's pamphlet goes on to say that when the above directions are followed, the leads and cut offs for each end of the cylinder for backward and forward gear are practically equal.

The arrangement of the gear just described is the most effective but considerable latitude is permissible. For instance the point C, can be placed above or below the center line of the connecting rod and the point K, can be raised or lowered, so that the line KO, is no longer parallel to AB, but it is not advisable that the line should have a greater inclination to AB, than 4° or thereabouts.

Again, the anchor link may be dispensed with, the point J, being guided in a slide affixed to some convenient part of the engine. For vertical engines the same rules apply by placing the diagram vertically and altering relatively the terms vertical and horizontal.

Joy and Bremme Gears Compared.—The Joy gear is preferred for locomotives, and the Bremme for marine engines.

It is somewhat difficult to arrange the Bremme gear with its eccentric rod and reversing arm underneath a locomotive boiler, so as to be compact, and to clear the various parts. In marine work, space for this is usually abundant. The movement of the parts of the Bremme gear is considerably less than the Joy.

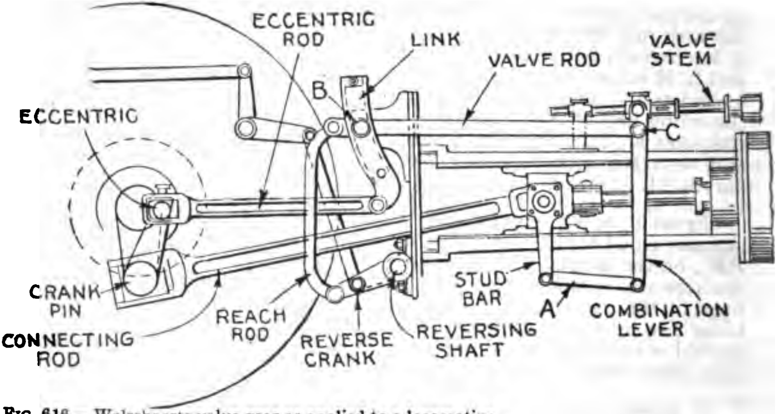

FIG. 616.—Walschaerts valve gear as applied to a locomotive.

Walschaert Gear.—This type of valve motion is one of the most important of the so called radial gears.

It was invented by Egide Walschaerts* (incorrectly spelled Walschaert, Walschart, etc.), of Mechlin, near Brussels, Belgium, and is especially adapted to locomotives.

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*NOTE.—***Egide Walschaerts*** died on the 18th of February, 1901, at Saint-Gilles, near Brussels, at the age of eighty-one years. His mechanism which is so original, has been adopted for many years in most of the countries of Europe ***and has been wrongly attributed*** to Mr. Huesinger von Waldegg. He was born January 21, 1820, at Malines, which place became, fifteen years later, the central point of the system of Belgian Railways. The line from Brussels to Malines was opened in 1835, and this event decided the career of young Walschaerts. Three years later, at the exhibition of products of Malines, there appeared some remarkable models executed by him, and described as follows in the catalogue: No. 19. M. E. Walschaerts, Jr., student of the Municipal College: (*a*) A stationary steam engine of iron (the main piston having the diameter of 4.5 cm. or 1.77 in.) (*b*) A working model of a locomotive in copper to the scale of $^{1}/_{20}$ of the railway locomotives. (*c*) Section of a stationary steam engine. (*d*) Model of a suction pump and a duplex pump. (*e*) Glass model of an inclined plane. Minister Rogier was so much struck by it that he had Walschaerts enter the University of Liege, but his studies were interrupted by a serious illness, and were never completed. We find traces of him at the National Exhibition in Brussels in 1841. The report of the jury mentions with praise a small locomotive constructed entirely by Walschaerts, and a steam boat 6.50 metres long and 1.75 metres wide, which was capable of carrying sixteen men and traveling (so the report says) at four leagues an hour on the canal. The boiler of this little boat was of a new system invented by the constructor. The jury does not give further details. Walschaerts received the silver medal. In 1842 Walschaerts was taken into the shops of the State Railway at Malines as a mechanic. Machine tools existed only in the most rudimentary forms, and the store rooms were badly provisioned. The lack of organization in the shops

The recent development of the locomotive in this country has presented conditions that has caused the extensive use of the Walschaerts gear in place of the shifting link.

The Walschaerts gear like other radial gears gives a constant lead and cannot be adjusted without disturbing the other events.

The layout of this gear is more or less a matter of trial, many minor locations may be varied in design, such as the position of the link pivot or the point where the eccentric rod is pivoted to the link, and in this way modifications in the action of the valve may be accomplished.

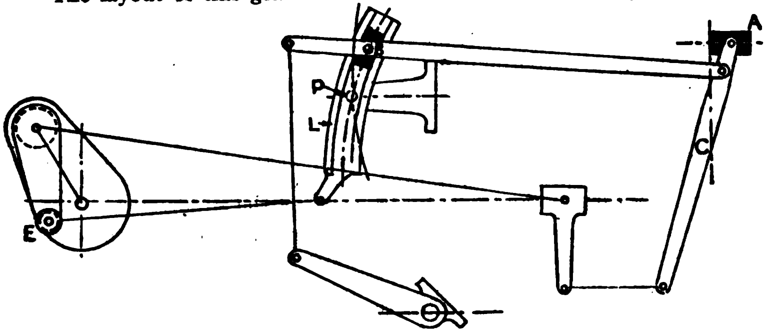

FIG. 617.—Skeleton diagram of Walschaerts valve gear. ***In operation,*** the movement given to the valve slide A, is the resultant of two components. The first is derived from the eccentric E, through the link L, and varies in amount as B, is moved out from P, and in direction relative to the crank as B is above or below P. The second component is derived from the cross head through the combination lever C. The resultant effect is equivalent to the motion that would be given by a single eccentric shifting along a straight line.

NOTE.—*Continued.*

rendered a man of Walschaerts abilities particularly valuable, and at the end of two years he was made shop foreman at Brussels. Although he was only twenty-four years of age he had already shown the qualities which make an engineer, which should have carried him in a few years to be the technical head of the motive power department. It is humiliating to be compelled to say that he remained shop foreman throughout his life. The first locomotive came from England and had not been in service for more than ten years when Walschaerts was made foreman. The railroad was growing rapidly and it was necessary to increase the forces and to acquire experience. Walschaerts was not content with the duties incurred in these difficult circumstances, but began his career by the invention of his system of valve motion. On October 5, 1844, Mr. Fischer, Engineer of the Belgian State Railways, filed for Egide Walschaerts an application for a patent relating to a new system of steam distribution applicable to stationary steam engines and to locomotives. The Belgian patent was issued on November 30, 1844, for a term of fifteen years. The rules of the department did not allow a foreman to exploit a Belgian patent for his own profit and this explains probably the intervention of Mr. Fischer, who has never claimed the slightest part, material or moral of the invention. On October 25th of the same year, Walschaerts took out a patent in France for the same invention. There also exists among the documents left by the inventor, a contract signed at Brussels in 1845 by Demeuldre, from which it appears that he undertook to obtain a patent of importation into Prussia for the new valve motion, subject to an assignment by Walschaerts of half of the profits to be deducted from the introduction of the new valve motion in this country. It is probable, however, that this contract was never carried out. The design attached to the Belgian patent is a primitive arrangement, the link oscillated on a fixed shaft, in regard to which it was symmetrical, but it had an enlarged

In the chapter on locomotives, the Walschaerts gear is described in detail, hence only an outline of its working principles need be given here.

The essential features of the gear are shown in fig. 617.

The motion of the valve is the resultant of two movements, one of which is intended to give constant lead, and the other, the required travel of the valve. These two movements are due to the cross head and eccentric, and are combined and imparted to the valve by the combination lever as shown in fig. 616.

Motion from the cross head is delivered to the lower end of the combination lever by a stud bar A, fixed to the cross head and rod B, giving it a reciprocating motion equal to the length of the stroke.

The second movement is transmitted from the eccentric through the eccentric rod, link pivoted at B, and valve rod (sometimes called radius rod), to the combination rod.

The valve rod is pivoted to the combination rod at a point C, near its upper end, so located that the motion received from the cross head will reciprocate the valve stem through a space equal to twice the linear advance, and thus to place the valve in position with constant lead at the beginning of the stroke.

The link is curved to a radius equal to the length of the valve rod. The valve rod has a block pivoted near its end, and arranged to slide in the link as shown.

The cut off is shortened, or motion reversed by shifting the block by means of the reverse crank and reach rod which joins this crank to the end of the valve rod.

NOTE.—*Continued.*

opening at the center so that only at the ends was it operated without play by the link block, which was made in the form of a simple pin. There was only one eccentric, the rod of which terminated in a short T, carrying two pins. The reverse shaft operated the eccentric rod and maintained it at the desired height. For one direction the lower pin of the T engaged in the lower end of the link, and to reverse the engine the rod was raised so that the upper pin engaged in the upper end of the link. The angle of oscillation of the link varied with the position of the pin in the link, and this oscillation was transmitted by an arm to the combining lever, which was also operated by the cross head. The central part of the link could not be used for the steam distribution, as it was necessary to enlarge it to allow for the play of the pin which was not in operation. It may be asked why the inventor used two separate pins mounted on a cross piece on the end of the eccentric rod, instead of a single pin on the center of the rod which would have served for both forward and backward motion without requiring the center enlargement of the link. It must be borne in mind that the raising or lowering of the eccentric rod by the reverse shaft was equivalent to a slight change in the angular advance of the eccentric. Consequently with a link of a sufficient length to keep down the effect of the angularity it was necessary to reduce as much as possible the movement of the eccentric rod. Notwithstanding its differences the mechanism described in the patent of 1844 is in principle similar to the valve motion with which every one is to-day familiar and which the inventor constructed as early as 1848, as is shown by a drawing taken from the records of the Brussels shops, on which appears the inscription "Variable expansion; E. Walschaerts system applied to Locomotive No. 98, Brussels, September 2, 1848."

CHAPTER 11

GOVERNORS

An important requirement of engine operation for most conditions of service is the maintenance of a practically constant speed under variable load. This control is accomplished by an automatic device called a *governor*. With respect to speed control, engines may be divided into two general classes, according as they are designed to run at

1. *Variable speed, or
2. Constant speed.

The speed regulation of the two types is classed respectively as

1. Hand control;
2. Automatic.

Under the first division are types such as marine, locomotive, and hoisting engines, while the second division consists of that large class known as stationary engines.

Classes of Governor.—The varied conditions of service give rise to numerous types of governor differing both in principle and construction. Accordingly, governors may be classed:

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*NOTE.—It should be understood that there is a type o. engine called "variable speed engine," which works under control of a governor so arranged that the speed may be altered as fully explained on page 398.

1. With respect to steam control as

   a. Throttling;
   b. Variable cut off.

2. With respect to the operating principle, as

   a. Centrifugal
      - gravity;
      - spring;

   b. Inertia.

3. With respect to operation, as

   a. Sensitive;
   b. Isochronous;
   c. Variable speed.

4. With respect to construction, as

   a. Pendulum
      - unloaded;
      - loaded.

   b. Shaft
      - shifting eccentric;
      - swinging eccentric;
      - double eccentric.

Principle of Centrifugal Governor.—The action of governors of the type depends upon *the change of centrifugal force when the rate of rotation changes.*

In these governors one of two resisting forces is employed as that due to gravity, or a spring. Gravity is usually the resisting force in pendulum governors, and one or more springs in shaft governors.

In fig. 618 let

h = height of cone of revolution;
r = radius of rotation of ball;
W = weight of ball;
C = centrifugal force due to speed of rotation

then

$$W \times r = C \times h$$

from which

$$W=\frac{Ch}{r};\quad C=\frac{Wr}{h};\quad r=\frac{Ch}{W};\quad h=\frac{Wr}{C}.$$

Ques. Upon what does *h*, or distance of the plane of the ball below the point of suspension (popularly expressed as "height of the ball") depend?

Ans. The distance *h* varies inversely as the square of the speed.

The weight of the ball and radius of revolution have no effect upon the position of the balls.

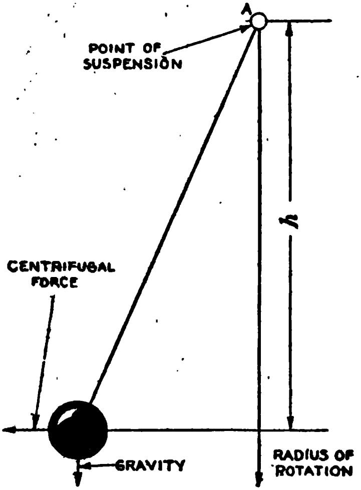


FIG. 618.—Simple revolving pendulum illustrating the principle of operations of centrifugal governors.

Pendulum Governors.—In its simplest form the pendulum governor consists of two balls suspended upon a vertical shaft as in fig. 619. The weight of the balls tends to hold them down, and centrifugal force operating against gravity (or a spring) tends to raise them (that is, make them fly outward), as explained in the preceding sections.

Theoretically, the action of the balls is independent of their weight as the centrifugal force varies in the same proportion as the weight and maintains constant the relative effects, so that at constant speed the balls will rotate in the same plane, whatever their weight. This is true only where the arms are simply hinged at the top without any other connections.

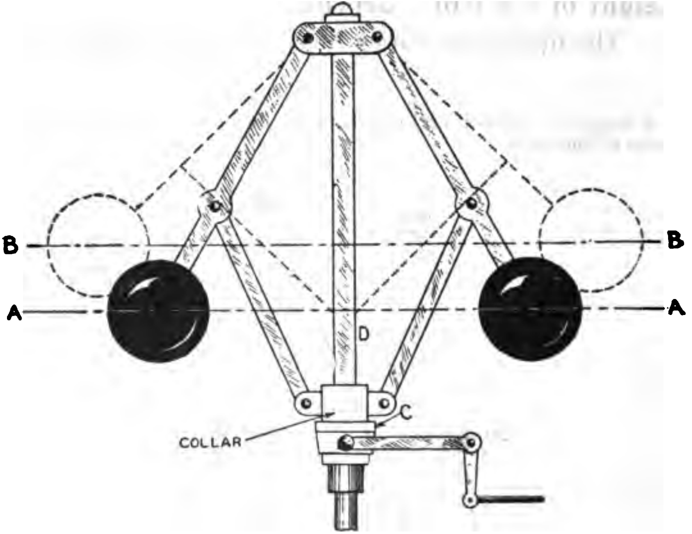
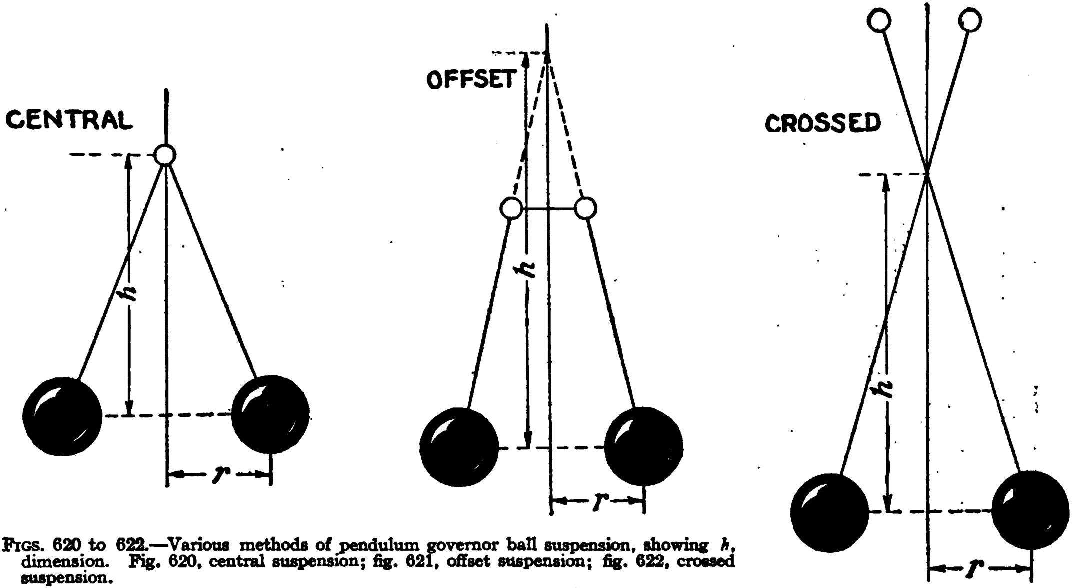


FIG. 619.—Simple pendulum governor actuated by centrifugal force and gravity. ***In operation***, assume that at normal speed of the engine, the balls revolve in the plane AA, then let part of the load be thrown off and the engine will speed up slightly, or enough to throw the balls into some higher plane as BB. This raises the collar on the spindle from C, to D, and by means of a lever attachment or equivalent, control the steam supply by throttling or variable cut off according as the governor is of the throttling or cut off type.

In a governor as constructed there are a collar, side rods, etc., which, having no rotary motion tending to raise them by centrifugal force, act as dead weights on the balls and cause them to revolve in a lower plane. However, if the balls be made heavy in comparison with the weight of the arms and collar, this effect becomes small.



FIGS. 620 to 622.—Various methods of pendulum governor ball suspension, showing *h*, dimension. Fig. 620, central suspension; fig. 621, offset suspension; fig. 622, crossed suspension.

In designing a governor the first step is naturally to assume an allowable speed variation and then compute *h* (figs. 620 to 622) for maximum and minimum speed. Then by knowing the rise and fall of the balls, the movement of the collar upon the stem can be determined, and the desired movement of the regulating device be obtained by interposing the proper gearing.

The expression for the total centrifugal force horizontally outward is:

$$C = 12\,W\,v^2 \div gr \tag{1}$$

in which

C = total centrifugal force in lbs.;
W = weight of both balls in lbs.;
v = tangential velocity of balls in feet per second;
g = acceleration due to gravity = 32.16;
r = radius of rotation of the balls, in inches.

In the previous section was obtained the expression

$$Wr = Ch \tag{2}$$

Substituting for C, its value as in (1),

$$Wr = 12\,W\,v^2\,h \div gr \tag{3}$$

from which

$$h = gr^2 \div 12\,v^2 \tag{4}$$

Now the tangential velocity v, of the balls for any rotative speed R, is

$$v = 2\,\pi\,\mathrm{r}\,R \div (12 \times 60) = \pi\,r\,R \div 360$$

and since $g = 32.16$, substituting in (4)

$$h = 32.16\,\mathrm{r}^2 \div 12\,(\pi\,r\,R \div 360)^2 = 35{,}191.7 \div R^2 \tag{5}$$

Example.—In an engine running at 200 r.p.m., the governor is to run at half that speed and have a "regulation" within 4 per cent.

Four per cent regulation means that the maximum speed of the engine is to be

$$200 \times 1.02 = 204 \text{ r. p. m}$$

and the minimum speed

$$200 \times .98 = 196 \text{ r. p. m.}$$

Then the corresponding maximum and minimum speeds of the governor will be half these values or 102 and .98 r. p. m. respectively. Substituting these values in (4), for maximum speed

$$h = 35{,}191.7 \div 102^2 = 3.38$$

and for minimum speed

$$h = 35{,}191.7 \div 98^2 = 3.66$$

which means that the balls must rise and fall vertically a distance of

$$3.66 - 3.38 = .28 \text{ inch}$$

for a total variation of 4 per cent in the speed of the engine, that is for 4 per cent regulation. The corresponding movement of the collar upon the governor shaft will depend upon the lengths of the different arms and connecting levers and may be determined graphically.

Loaded Pendulum Governors.—In the example given in the preceding section it is evident that there is but little change in the height of the governor for even considerable variation in speed, and also that for high speeds, the height h of the governor becomes so small that the mechanism would be difficult to construct. To overcome these defects the governor may be *loaded*, that is, a weight is placed on the collar to assist gravity in holding down the balls as in fig. 623.

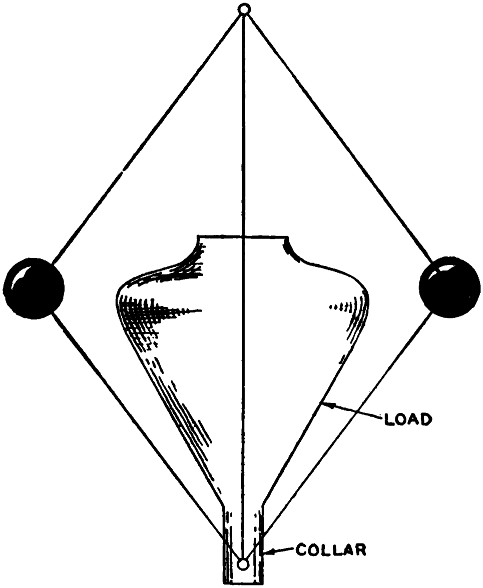


FIG. 623.—Diagram of loaded pendulum governor. In this type a weight (or equivalent) is placed on the collar, thus increasing the gravity effect in resisting the centrifugal force.

The governor equation already found may be stated as

gravity moment = centrifugal force moment

or in symbols

$$Wr = Ch \quad \ldots\ldots\ldots\ldots \quad (6)$$

Now if W, be the combined weight of the balls, as before, and W′ the weight of the load placed upon the collar as in fig. 623, then the gravity moment is (W+W′) r, which substituted for W in equation (6) gives

$$(W+W')\,r=Ch \qquad (7)$$

Substituting in (7) the value of C, as found in (1)

$$(W+W')\,r=12\,W\,v^2\,h\div gr$$

from which, solving for h

$$h=\frac{W+W'}{W}\times\frac{gr^2}{12v^2} \qquad (8)$$

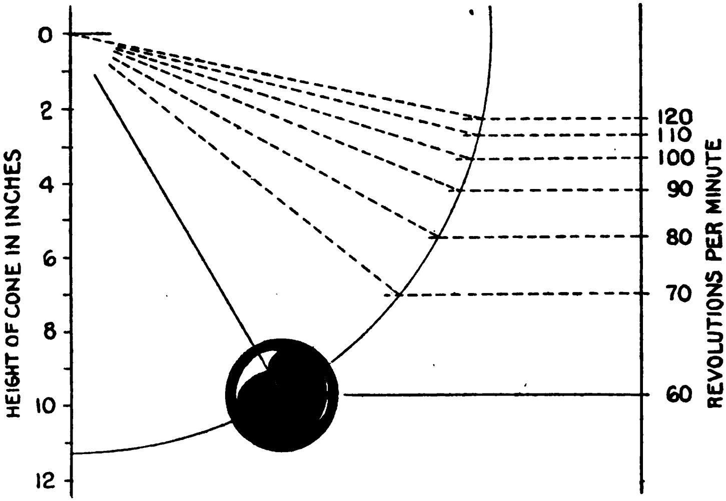


FIG. 624.—***Reason*** for loading a governor. If the height h, for a simple pendulum governor be calculated for different speeds, it will be found that while the change of height h, for a change of speed may be comparatively large when the speed of the governor is low, when the speed is increased, the change of height for a change of speed becomes so small as to be of no practical value. Thus in the figure from, say 60 to 70 revolutions, the difference in h, is large, and say from 110 to 120, it is very small. Hence if a weight or load be added which increases the gravity effect, then a large movement of the sleeve may be obtained with a high speed of rotation, and at the same time a much more powerful governor is obtained than when no central load is used.

It will be noticed that the second factor of (8) is the same as the value of h, in equation (4), and reduced in (5), hence substituting the value given in (5) in equation (8).

$$h=\frac{W+W'}{W}\times\frac{35{,}191.7}{R^2}$$

from which it appears that the height h, of a loaded governor is greater than an unloaded one to the extent of the factor $(W+W')\div W$.

Assuming the load upon the collar to be 1, then for maximum speed

$$h=\frac{1+5}{1}\times\frac{35{,}191.7}{102^2}=6\times3.38=20.28$$

and for minimum speed

$$h=\frac{1+5}{1}\times\frac{35{,}191.7}{98^2}=6\times3.66=21.96$$

Then the vertical movement of the collar will be 21.96 — 20.28 = 1.68 in., as compared with .28 in the example of the unloaded governor just given.

Ques. How is the load proportioned?

Ans. It is usually made much heavier than the balls.

Ques. What equivalent is sometimes used in place of a weight on loaded governors?

Ans. Springs.

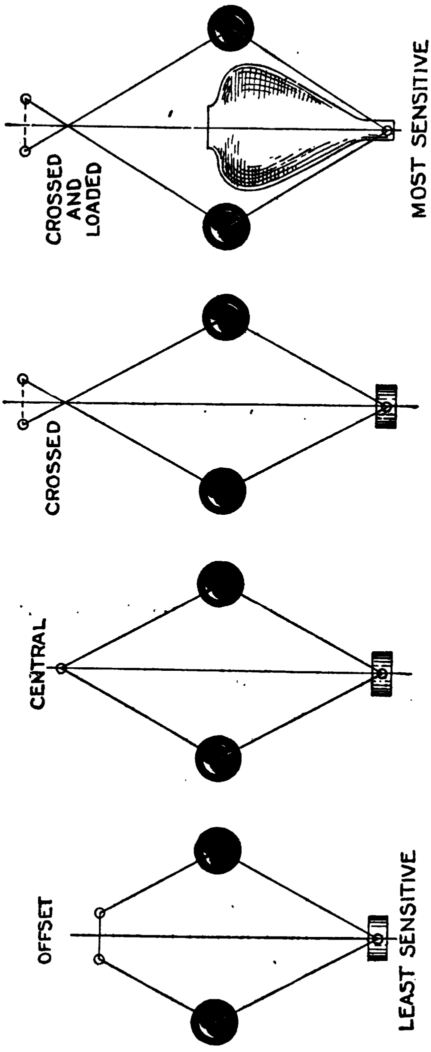
These are used especially on throttling governors and are so attached that they oppose the centrifugal force obtaining the same effect as a weight.

Ques. How may a loaded governor be constructed and why?

Ans. It may have comparatively small rotating weights, because the centrifugal force increases as the square of the number of revolutions and only directly as the weight of the balls.

Sensitiveness.—A governor is said to be sensitive *when a small variation of speed causes a considerable movement of the regulating mechanism.*

Theoretically, loading a governor has no effect on its sensitiveness however since, in practice, the friction of the governor and regulating mechanism may be considerable, the sensitiveness of a loaded governor is actually much greater than that of the unloaded type.



FIGS. 625 to 628.—Pendulum governors arranged according to their degree of sensitiveness, fig. 625 being the least sensitive and fig. 628, the most sensitive.

About 2 per cent variation of speed of the engine may be considered as the practical limit of variation with good governors. A less percentage than this requires an abnormally large fly wheel.

Figs. 625 to 628 show several types of pendulum governors arranged in the order of their sensitiveness, fig. 625 being the least sensitive, and fig. 628 the most sensitive.

Stability.—A governor is said to be stable *when it maintains a definite position of equilibrium at a given speed.*

When the reverse conditions obtain, it is said to be *unstable*, that is, when at a given speed it assumes indifferently any position throughout its range of movement.

Ques. What is the condition for stability?

Ans. For stability, *the centrifugal force must increase more rapidly than the radius of rotation of the balls.*

Evidently no governor can maintain a constant speed since it requires a change of speed to actuate the regulating mechanism. When the balls are in the lowest position the regulating mechanism gives the full steam supply for maximum load and when the balls are highest, just enough to run the engine against its frictional load.

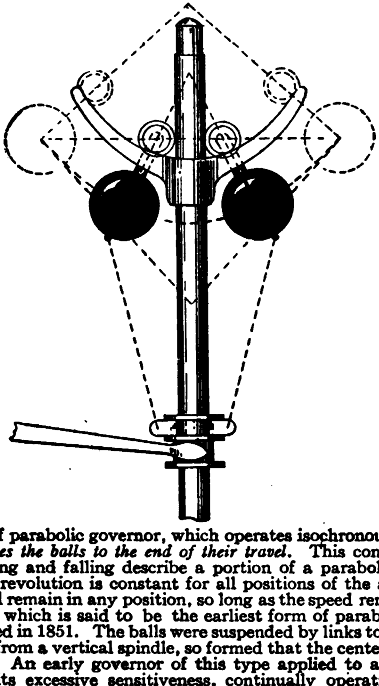


Fig. 629.—Early form of parabolic governor, which operates isochronously, that is, *the slightest variation of speed drives the balls to the end of their travel.* This condition is only obtained when the balls in rising and falling describe a portion of a parabola, for, in this case the height of the cone of revolution is constant for all positions of the arms, and the balls are in equilibrium and will remain in any position, so long as the speed remains unchanged. ***The above construction*** which is said to be the earliest form of parabolic governor known in England was introduced in 1851. The balls were suspended by links to rollers, which traveled upon arms branching from a vertical spindle, so formed that the centers of the rollers traveled in a parabolic curve. An early governor of this type applied to a compound engine was rendered useless by its excessive sensitiveness, continually operating the throttle valve. The difficulty was overcome by applying an air dash pot.

If the boiler pressure or the load be changed, a certain amount of displacement of the balls is necessary to vary the steam supply, and this displacement can only be obtained by a change in speed, hence the term constant speed is erroneously used as applied to engines where speed is controlled by a governor.

Isochronism.—A governor is said to be **isochronous** *when it is in equilibrium at only one speed.*

If, when the balls are displaced, the centrifugal force changes proportionately to the radius of rotation of the balls, the speed is constant, that is, the equilibrium of the governor is neutral, allowing it to revolve in equilibrium at only one speed.

The slightest variation in speed drives the balls to the end of the travel. Such a governor is said to be *isochronous*, and its sensitiveness is theoretically infinitely great.

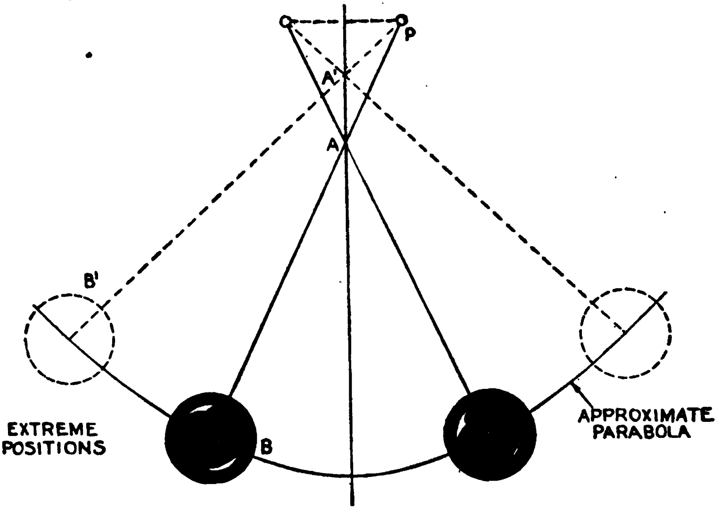


FIG. 630.—Diagram of crossed arm pendulum governor, the approximate equivalent of the parabolic type. By producing the two extreme positions of an arm BA and B′A′, until they intersect at P, and using this as a point of suspension. The curve described by the balls during their travel will approximate a parabola and the action of the governor will be approximately isochronous.

Ques. What type of governor is isochronous?

Ans. The parabolic governor.

Hunting.—An isochronous governor cannot be used successfully on an engine without being modified so as to obtain a small margin of stability to prevent violent changes in the steam supply, especially if there be much frictional resistance to be overcome by the governor, or where the engine responds slowly to the influence of the governor. When a change of speed occurs, however quickly the governor acts, the engine's response is more or less delayed.

If the regulation be by throttling, the steam chest forms a reservoir to draw upon, and if by variable cut off, the opportunity is lost if cut off has already occurred, and control cannot begin until the next stroke. A sudden decrease of load is accompanied by such increase in speed as to cause abnormal governor action giving too little steam for the reduced load. Causing a decrease of speed accompanied by excessive increase in the steam supply. The governor thus oscillates in its endeavor to find a position of equilibrium and such action brought on by over-sensitiveness is called *hunting*.

Ques. What form of governor is approximately the equivalent of a parabolic governor?

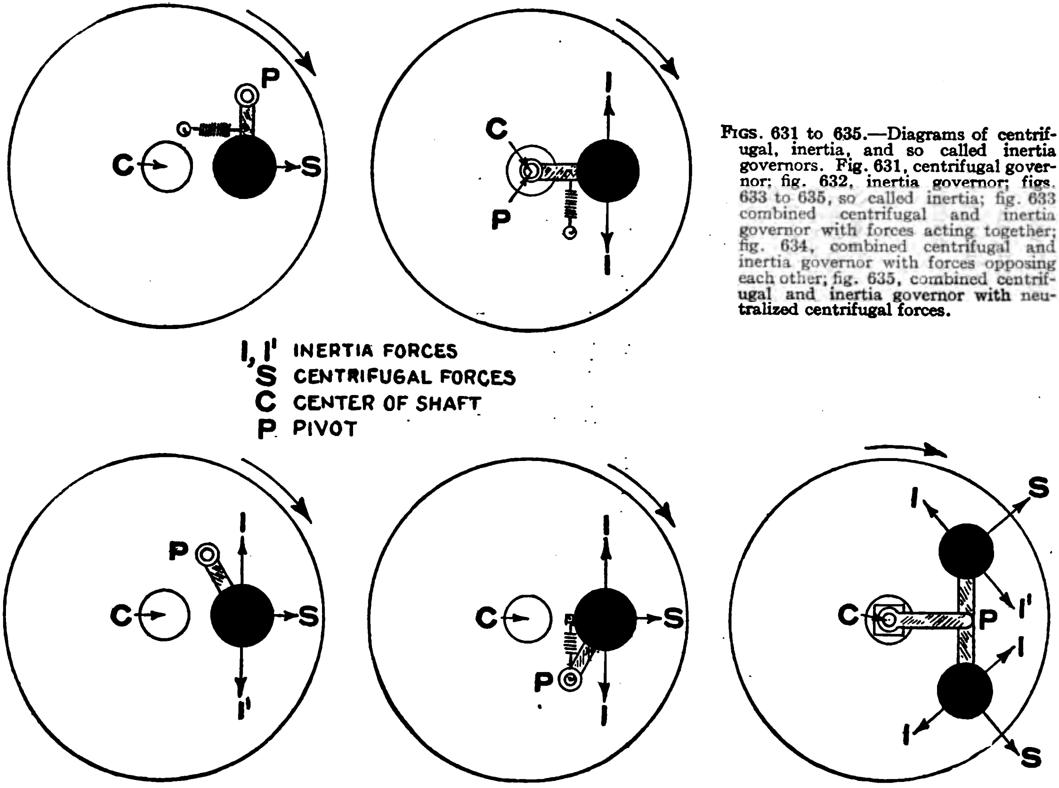
Ans. The pendulum governor with crossed arms as in fig. 630.

Ques. How is the stability of this governor varied?

Ans. Reducing the distance P A, increases the stability; increasing P A, gives the reverse effect. If the point P, be too far from the spindle, the height of the cone may increase as the balls rise and cause unsatisfactory operation.

Inertia Governors.—The term *inertia* may be defined as *that property of a body by virtue of which it tends to continue in a state of rest or motion in which it may be placed until acted upon by some force.* This forms the working principle of inertia governors, and in general an inertia governor may be said to consist of a heavy body pivoted at its center of gravity and connected at a proper point to the regulating mechanism.

In operation, the heavy body revolving in step with the fly wheel tends to revolve at constant speed. Any change of speed of the fly wheel due to



FIGS. 631 to 635.—Diagrams of centrifugal, inertia, and so called inertia governors. Fig. 631, centrifugal governor; fig. 632, inertia governor; figs. 633 to 635, so called inertia; fig. 633 combined centrifugal and inertia governor with forces acting together; fig. 634, combined centrifugal and inertia governor with forces opposing each other; fig. 635, combined centrifugal and inertia governor with neutralized centrifugal forces.

variable load produces a change in the position of the heavy body with respect to the wheel, thus moving the regulating mechanism.

The so called inertia governors found on many are, strictly speaking, combined centrifugal and inertia governors, as their action depends on both forces. According to constructing, one force may be made to either oppose or assist the other.

In fig. 631 a governor disc is pivoted to the fly wheel at P, as shown. If the wheel rotate in the direction of the arrow, an increase in speed will cause the disc to move outward from the center C, by centrifugal force, but if the governor disc be pivoted at the center of the shaft as in fig. 632, then the centrifugal force acting radially with respect to the pivot P, will have no effect on the disc since the position of the pivot P, coincides with the center C, of the fly wheel.

If in fig. 632 the speed of the fly wheel increase or decrease, the disc, (tending to rotate at constant speed) will respectively lag behind or advance beyond the position shown in the figure, that is, it will move with respect to the fly wheel in the direction of I, or I′, respectively, inertia in this case alone being the controlling force, the resisting force being the tension of the spring.

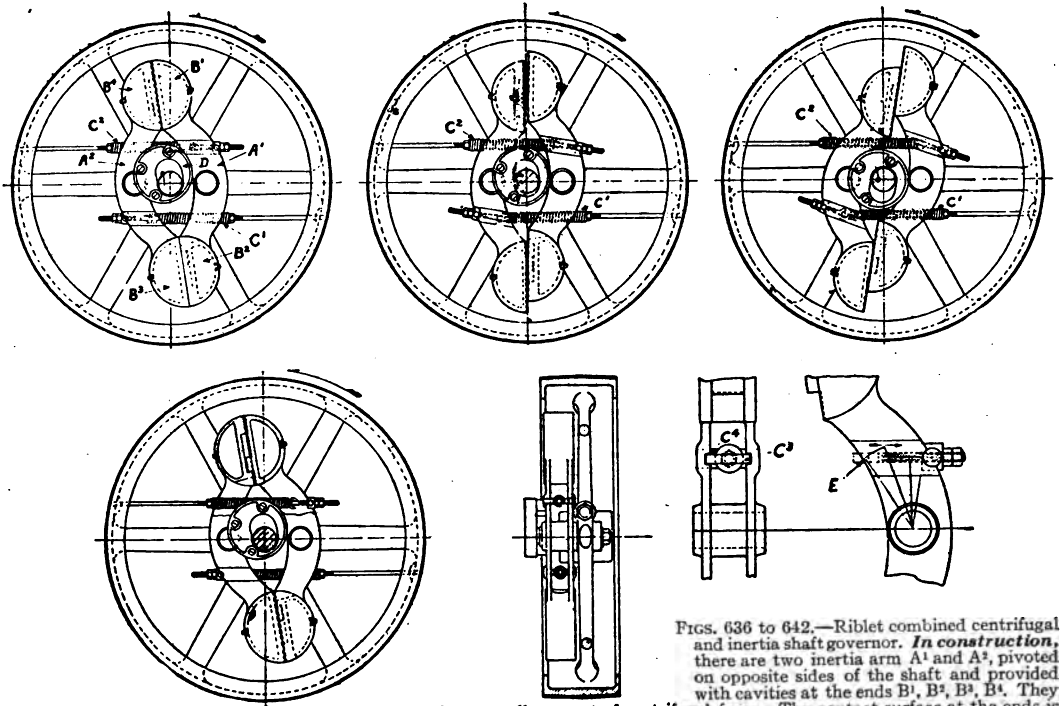
In fig. 632 the force due to inertia is a maximum. This force may be made to assist or oppose the centrifugal force according to the location of the pivot P. Thus in fig. 633, if the fly wheel be rotating in the direction of the arrow, both forces act together and make the governor rapid in its movement, while in fig. 634, the forces oppose each other and tend to make the governor sluggish.

Another form is shown in fig. 635, in which the centrifugal force is neutralized, the controlling force being due to inertia alone. In the figure two discs A and B, connected by an arm are pivoted at the center of gravity P, at a distance C P, from the center C, of the fly wheel.

In operation when the fly wheel revolves disc A, is acted upon by a centrifugal force F, and B, by an equal centrifugal force F′. These forces acting on opposite sides of the pivot P, and at equal distances neutralize each other.

Now if the fly wheel revolve in the direction of the arrow, and its speed be decreased, the inertia of the discs (represented by the forces I′ I′) will cause them to revolve around P, to some position as A′ B′. Clearly if the speed increase the reverse condition will obtain, that is, the discs will revolve around P, to some position as A″ B″, under the influence of the inertia forces I′ I′.

Spring Governors.—The use of springs for the controlling force or to assist gravity is quite common. When springs are used the ball arms may be arranged to travel across a vertical



FIGS. 636 to 642.—Riblet combined centrifugal and inertia shaft governor. ***In construction,*** there are two inertia arm A^1 and A^2, pivoted on opposite sides of the shaft and provided with cavities at the ends B^1, B^2, B^3, B^4. They are pivoted slightly out of the center of gravity to give a small amount of centrifugal force. The contact surface at the ends is not at right angles with the suspension pins, but at the correct angle to allow the arms to swing a sufficient amount to give the eccentric its maximum and minimum throw. The arms do not come in contact at both ends except at either extreme of travel (figs. 636 and 638), and spread somewhat (fig. 637), in all intermediate positions. The arms are held in contact at one end, axis, or the governor may be operated in a horizontal position and gravity practically eliminated.

Spring governors can be made practically isochronous if desired, by so adjusting the spring that the initial compression in the spring bears the same ratio to the total compression that the minimum radius of the balls bears to the maximum radius.

In practice, stability is provided by making the spring a little stronger than the above adjustment.

Fig. 692 illustrates the operation of a spring governor. As shown, the balls are attached to bell crank arms, pivoted at P and P', to a frame, which revolves around the central upright shaft. The travel of the balls is transmitted by the bell cranks to the collar at A A'. An adjustable spring presses against the collar and acts as a substitute for gravity.

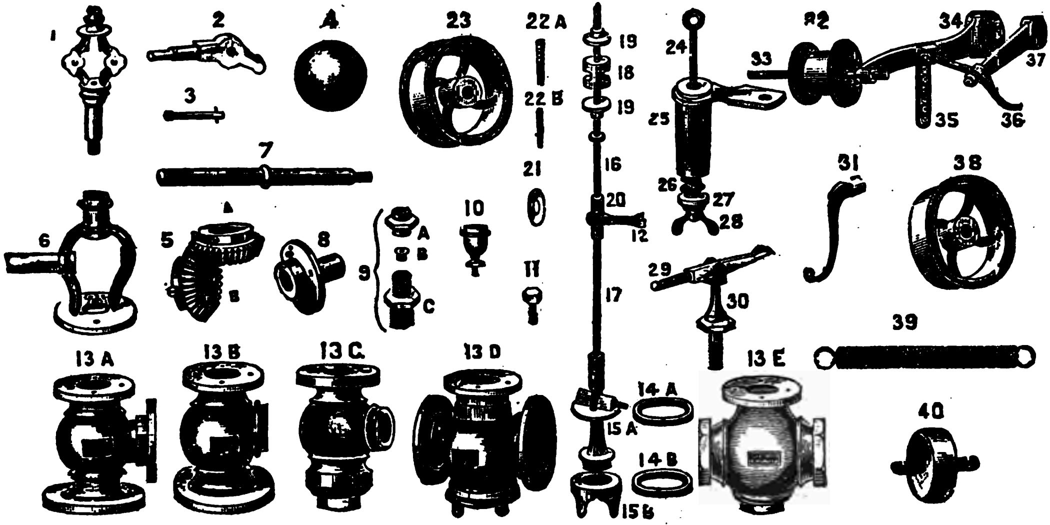
In operation as the speed increases the centrifugal force increases and the balls move outward. The compression on the spring increases similarly, and by suitably adjusting the initial pressure on the spring, the governor may be made nearly isochronous, as before mentioned, and rendered very sensitive, which is a characteristic of this class of governor.

The Regulating Mechanism.—This term is applied to the gearing or "transmission" which transmits the movement of the governor to the control device.

FIGS. 636 to 642.—*Continued.*

(fig. 637), by having more tension on the eccentric arm spring C^1, than is maintained on the spring attached to the other arm C^2. By this arrangement any amount of friction necessary for stability is obtained. The eccentric D, is fastened to one arm A^2, in its proper relation to the suspension and crank pin so that the swinging of the arms move the eccentric across the shaft, changing its throw, producing the desired effect of changing the cut off of the valve. The sliding block, C^3, to which the arm ends of the springs are attached is used to increase the range of the springs. ***In adjusting,*** by moving the sliding block along the grooves C^4, the points of suspension are changed. By changing the points of suspension any desired effect upon the action of the governor is obtained. The inertia arms are purposely made heavy so as to balance the reciprocating parts of the valve motion.

NOTE.—Directions for Riblet combined centrifugal and inertia governor. 1. When springs are placed in the governor, the suspension points E, of the sliding block must be about half way between their minimum and maximum positions. 2. The free arm spring C^2, must be tightened just enough to hold it in place. 3. The eccentric arm spring C^1, must be tightened so that it is pulled out about ¾", or just enough to stop the rattle of the governor. 4. Start the engine. 5. If the engine run too fast, put shot in cavity B^1, and the same amount in B^3. 6. If the engine run too slow, put shot in cavities B^2 and B^4. 7. Until the point is reached where the required speed is obtained pay no attention to regulation. When the speed is obtained apply load to the engine. 8. If the engine run slower as the load is applied, move the springs by means of the sliding block toward their maximum position in direction of arm, fig. 642 of their points of suspension E; continue this movement until the desired regulation is reached. 9. If engine run faster as the load is applied, reverse the procedure given in 8. 10. Be sure the governor does not stick or bind and that the suspension pins are well lubricated with good grease.



FIGS. 643 to 691.—Parts of the Judson throttling governor. 1, head; 2, arms (2); 3, arm bolts (2); 4, balls (2) plain; 5, A, gear, upper, B, gear, lower; 6, arch; 7, pulley shaft; 8, pulley flange; 9, stuffing box, complete, A, stuffing box, cap, B, stuffing box, follower, C, stuffing box, bottom; 10, oil cup; 11, cap screws for arch; 12, band stop; 13, A, valve chamber, flanged, base and side, B, valve chamber, flanged, base, side and screwed, C, valve chamber, screwed base and side, D, valve chamber, horizontal flanged, E, valve chamber, horizontal screwed; 14, A, valve chamber, seat, upper, B, valve chamber, seat, lower; 15, A, valve chamber, piston, upper part, B, valve chamber, piston, lower part; 16, steel valve rod; 17, brass valve rod; 18, spool; 19, collar (2); 20, union; 21, washers; 22, A, taper pins for gears (2), B, taper pins for connecting rod (5); 23, pulley; 24, speeder, stem with eye; 25, speeder, barrel; 26, speeder, spring; 27, speeder, follower; 28, speeder, thumb nut; 29, speeder, lever; 30, speeder, standard and cap screw; 31, speeder sawyer's lever; 32, stop motion idler pulley; 33, stop motion idler pulley shaft; 34, stop motion drop arm; 35, stop motion catch; 36, stop motion cross lever; 37, stop motion cross lever arm; 38, pulley; 39, spring governor side springs (2); 40, spring governor ball with flat sides (2).

According to the nature of the regulations governors are classed as

1. Throttling;
2. Cut off.

The throttle valve as introduced by Watt was what is now known as a butterfly valve, and consisted of a disc turning on a transverse axis across the center of the steam pipe. It is now usually a globe, gate, or piston valve.

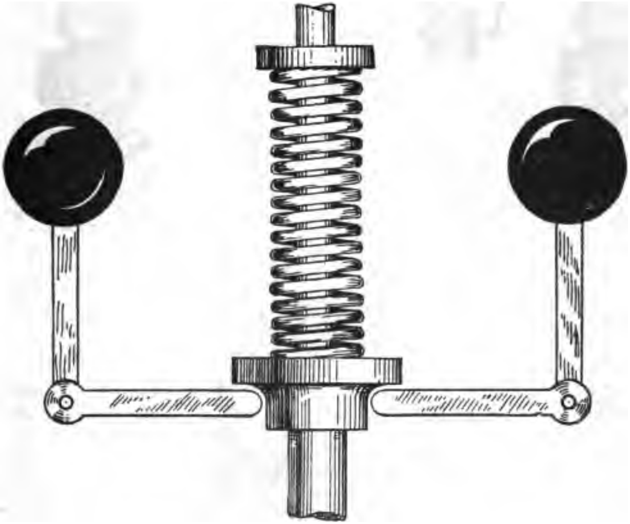


FIG. 692.—Elementary diagram showing working principle of spring governors.

When regulation is effected by varying the cut off, an expansion valve or the slide valve or piston type is used, the governor generally acts by changing the travel of the valve. In some forms of automatic expansion gear, the lap of the valve is altered; in others, the governor acts by rotating the expansion valve eccentric on the shaft and so changing the angular advance. These matters are fully explained in the chapters on valve gears.

Throttling Governors.—If this type of governor had never been invented, no doubt some of the world's natural resources

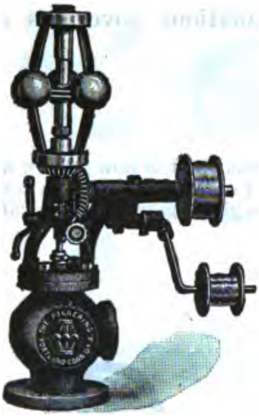


Fig. 693.—Pickering class A throttling governor fitted with automatic safety stop, speed regulator and sawyer's lever.

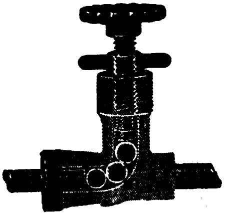


Fig. 695.—Pickering "ball speed ranger," permits increasing speed of engine 50 to 75 per cent from normal by turning the small hand wheel, which can be done while the engine is running.

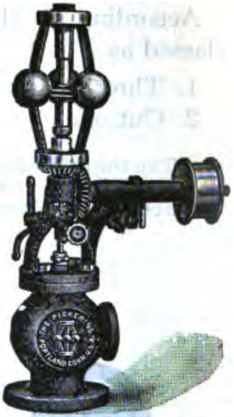


Fig. 694.—Pickering class B throttling governor fitted with speed regulator and sawyer's lever.

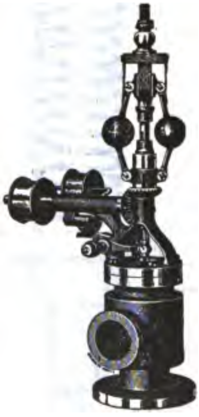
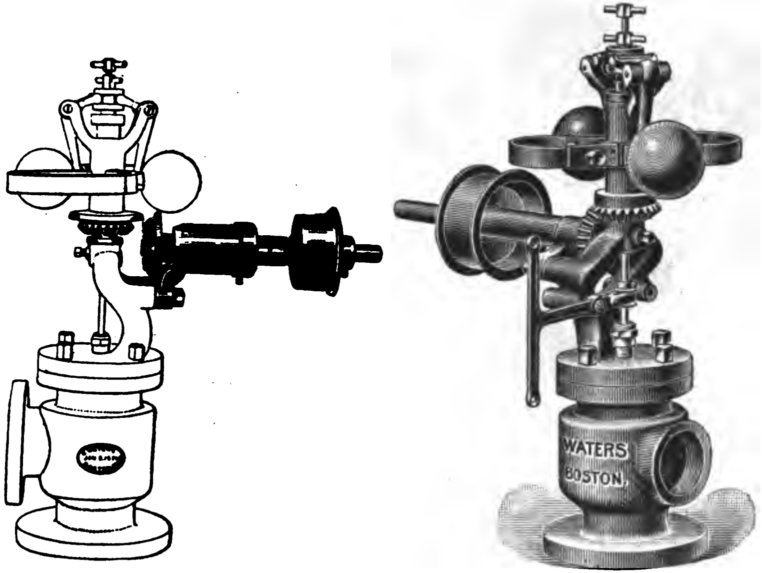


Fig. 696.—Sinker-Davis "Hoosier" throttling governor with wide range speed regulator permitting variation of 150 revolutions. ***Adjustment:*** Adjust coil springs only to the point where the valve works freely and easily. Only sufficient tension is required to balance the valve against the steam resistance. The coil springs are not for speed regulation. ***To change the speed:*** Loosen the lock nuts at the top of the governor above the traveling head, and if engine speed is to be increased, run the nuts down until the speed increases to the revolutions wanted, then lock them. If the speed is to be decreased, run the nuts up until the speed decreases to the revolutions wanted, then lock them. The regular speed of the governor is stamped on the traveling head, and a variation of speed not exceeding 75 revolutions, slower or faster, can be obtained by changing the nuts as explained above. If more variation be required, change the pulley on the main shaft of engine to give required speed on engine and on governor. The cam or automatic stop is adjustable for either right or left hand engines. The governor should be well oiled before starting, but can be oiled while in motion by oiling above the traveling head only; the oil will work down through the governor, and it is not necessary to oil at any other point. In packing the stuffing box, be sure to see that ***the valve stem works freely and without tension on the coil springs.***

would be better preserved and the price of fuel not so high. However, for some services where waste material is to be disposed of, and can be used as fuel, a throttling governor may be employed. This type of governor may be defined as **an automatic throttle valve** *which governs by altering the pressure at which steam is admitted to the cylinder;* that is, the throttle valve is opened or closed inversely with changes in load, thus causing more or less drop in pressure so that the resulting mean effective pressure in the cylinder will vary with the load and maintain a steady speed.



Figs. 697 and 698.—**Waters spring throttling governor. Fig. 697 shows class A, fitted with automatic safety stop in which a spring throws the shaft out of gear when the governor belt breaks. *In erecting,* be sure and have the end cap on the bracket stand with the *head of set screw pointing direct to the pulley on engine shaft. In operation,* if the governor belt break, the spring throws the shaft out of gear, the top drops and closes the valve. To start again, raise the top part, push the gears together and hold it in position while putting on the belt. Fig. 698 shows governor fitted with sawyer's lever.**

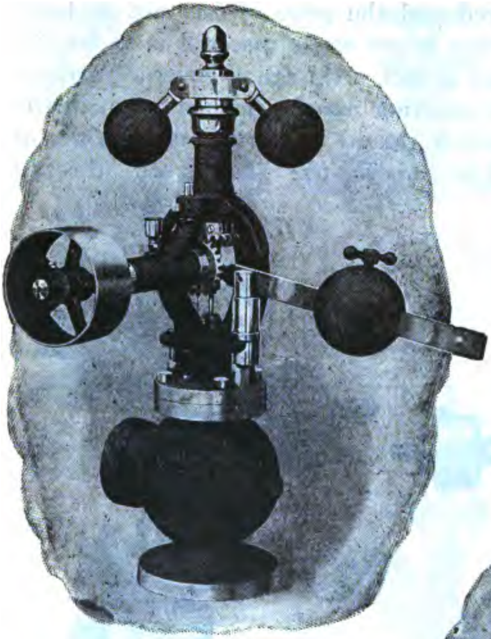


FIG. 699.—Gardner class A throttling governor fitted with automatic safety stop and speed regulator, sizes 1¼ to 16 inches inclusive. Class A type is of the gravity action and is especially adapted for the larger types of stationary engines. ***In operation*** the centrifugal force of the balls is opposed by the resistance of a weighted lever and the speed is varied by the position of the weight on the lever. The automatic safety stop is accomplished by permitting a slight oscillation of the shaft bearing, which is supported between centers and held in position by the pull of the belt; a projection at the lower part of the shaft bearing supports the fulcrum of the speed lever. If the belt break or slip off the pulley, the support of the fulcrum is forced back, allowing the fulcrum to drop, closing the valve. The valve chamber is fitted with valve seats made of a composition. The valve is of the same material. This style is made for both horizontal and vertical engines.

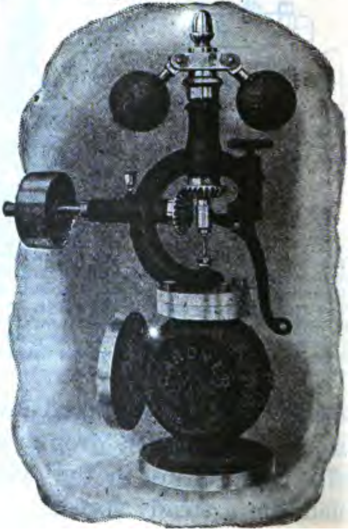


FIG. 700.—Gardner class B throttling governor fitted with speed regulator and Sawyer's lever. Class B combines both the spring and gravity actions, adapted for all styles of slow and medium-speed stationary engines. ***In operation*** the centrifugal force of the balls operates against the resistance of a coiled steel spring enclosed within a case and pivoted on the speed lever; by means of a screw the amount of compression on the spring can be changed so as to give a wide range of speed. A continuation of the speed lever makes a convenient sawyer's hand lever. By attaching a cord to this lever the valve of the governor can be controlled at a reasonable distance from the governor. Sizes from ¾ inch to 1¼ inches, inclusive, have swivel frames which can be set at any desired angle in relation to valve chambers. The valves and seats of this style are the same as used on the Standard Class "A" governor.

Fig. 702 shows the effect of a throttling governor upon the indicator diagram.

When working under full load, the diagram has the form shown by the full line, but when the load drops, and the engine speeds up slightly, the governor acts, partially closes the throttle valve, and the pressure is reduced

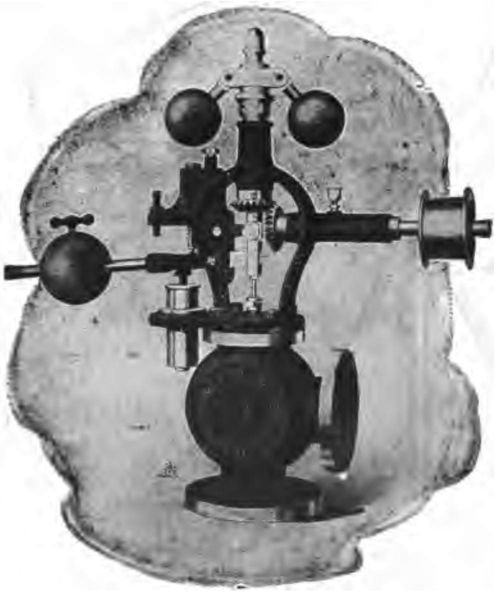


FIG. 701.—Gardner speed and pressure regulator fitted with automatic stop for steam actuated compressors. This regulator consists of a class A governor fitted with a brass cylinder containing a piston upon which the air pressure is exerted. The brass cylinder is connected by a pipe with the air receiver set about 25 feet distant from the regulator, so that the latter may always be under the direct influence of the air pressure within the receiver. The air pressure to be maintained is regulated by the position of the weight on the lever. When this pressure has been reached it is exerted on the brass piston, pushing it upward and closing the governing valve, or keeping it open just wide enough to maintain a constant air pressure. On duplex machines, when the desired air pressure has been attained, the regulator will bring compressor to a dead stop, starting it up automatically when air pressure falls below the required amount. On single compressors it is not desirable to bring the compressor to a dead stop, and there is an adjustable device on the regulator which, when set for certain pressure, will allow the compressor to just turn over when that pressure has been attained. The standard or ball governor acts merely as a speed controller; it has no throttling action on the steam until the limit of speed has been reached. By the use of properly proportioned pulleys on the governor and compressor, provision can be made for the proper speed limit. The ball governor keeps the compressor from exceeding this limit, and it thus serves to prevent the engine running away in case of sudden loss of air pressure from any cause.

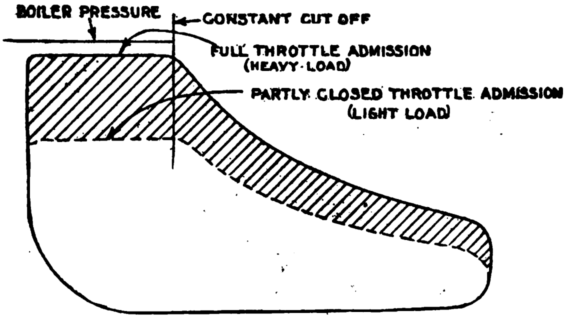


FIG. 702.—Indicator diagram showing action of throttling governor.

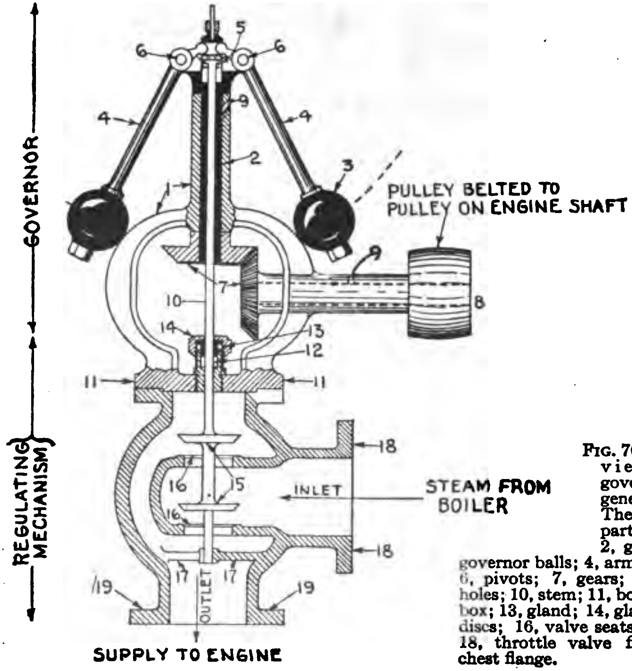


FIG. 703.—Sectional view of throttling governor showing general construction. The names of the parts are: 1, standard; 2, governor shaft; 3, governor balls; 4, arms; 5, stem swivel; 6, pivots; 7, gears; 8, pulley; 9, oil holes; 10, stem; 11, bonnet; 12, stuffing box; 13, gland; 14, gland box; 15, valve discs; 16, valve seats; 17, stem guard; 18, throttle valve flange; 19, valve chest flange.

by wire drawing so that the admission and expansion lines take the positions shown by the broken lines in the illustration. The resulting drop in pressure is always proportional to the reduction in load, so that the speed remains constant, or practically constant within certain limits, whatever the load upon the engine.

The general construction of a throttling governor is shown in fig. 703.

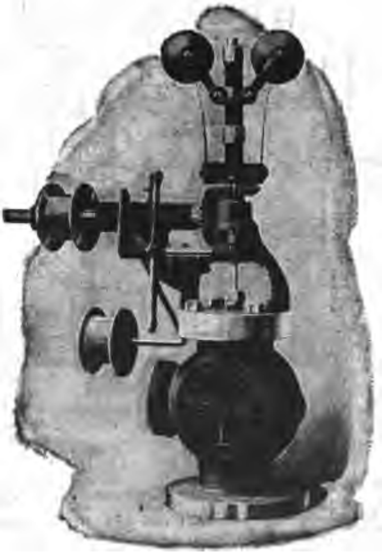


Fig. 704.—Gardner spring throttling governor with speed regulator, sawyer's lever, and automatic safety stop. This governor is recommended for traction and high speed stationary engines. It is very quick and sensitive in action, and is therefore capable of responding promptly to the various changes in load. The balls are rigidly connected to steel springs, the lower ends of the springs being secured to a revolving sleeve which receives its rotation through mitre gears; links connect the balls to an upper revolving sleeve, which is free to move perpendicularly. The balls at the free ends of the springs furnish the centripetal force, and the springs are the main centripetal agents. No gravity is employed. Sizes, ½ to 7 inches.

The valve 15 is of the balanced or double seat poppet type.

A different type of valve is shown in fig. 705 which is a sectional view of the regulating mechanism of the Waters throttling governor.

Cut Off Governors.—In this method of regulation, which is always used where any regard is had for economy, *steam is admitted with full throttle opening and the mean effective pressure controlled to suit the load by varying the point of cut off.*

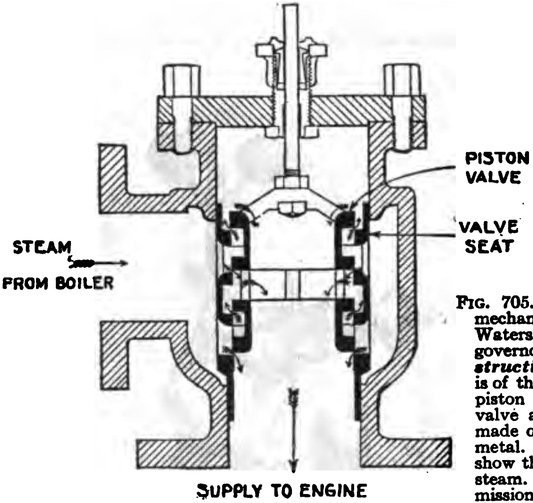


Fig. 705.—Regulating mechanism of the Waters throttling governor. *In construction,* the valve is of the triple ported piston type, both valve and seat being made of composition metal. The arrows show the paths of the steam. Ample admission area is secured by the three ports. The four valve seats are all in one casting, which fits the iron body at the ends only, providing for expansion and contraction. The valve being of the piston type is balanced and the triple ports give the required admission with a correspondingly small travel.

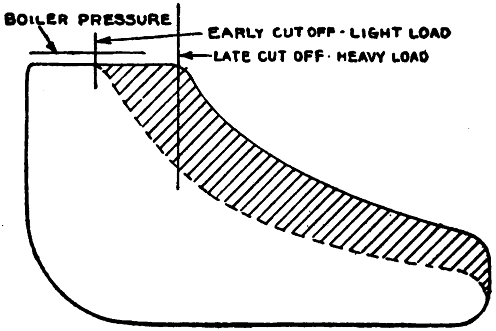


Fig. 706.—Indicator diagram illustrating regulation by variable cut off.

Fig. 706 shows the resulting changes upon the indicator diagram. Here the initial pressure remains the same, but the area of the card, and consequently the amount of work, is reduced by shortening the cut off. The original and final areas of the diagrams are the same in each case, and the reduction in work per stroke is shown by the shaded area.

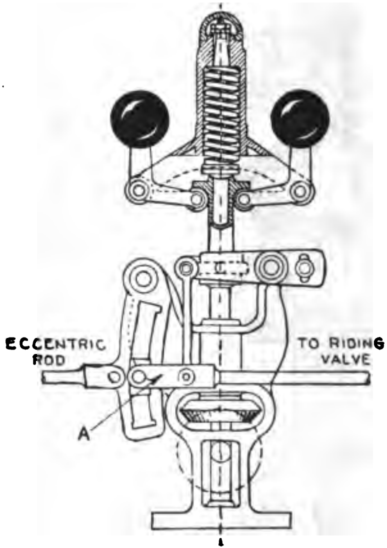


FIG. 707.—Hartwell spring ball governor with variable cut off regulating mechanism. *In operation,* as the speed increases, the governor raises the position of lever A, and the travel of the valve is reduced. This governor is capable of very close regulation, and when the speed exceeds a given number of revolutions, the steam supply may be entirely cut off.

In this class of governor, variable cut off is effected in several ways, as by

1. Variable travel of the valve;
2. Variable angular advance;
3. Combined variable travel and variable angular advance.

Fig. 707 shows a spring ball governor with regulating mechanism for varying the cut off by the first method. As shown, the regulating mechanism regulates the travel of a riding cut off valve by the movement of the lever A, in the slotted link B.

Regulation.—The term regulation means *speed variation*, usually expressed as a percentage of the normal speed of an engine, running under control of a governor.

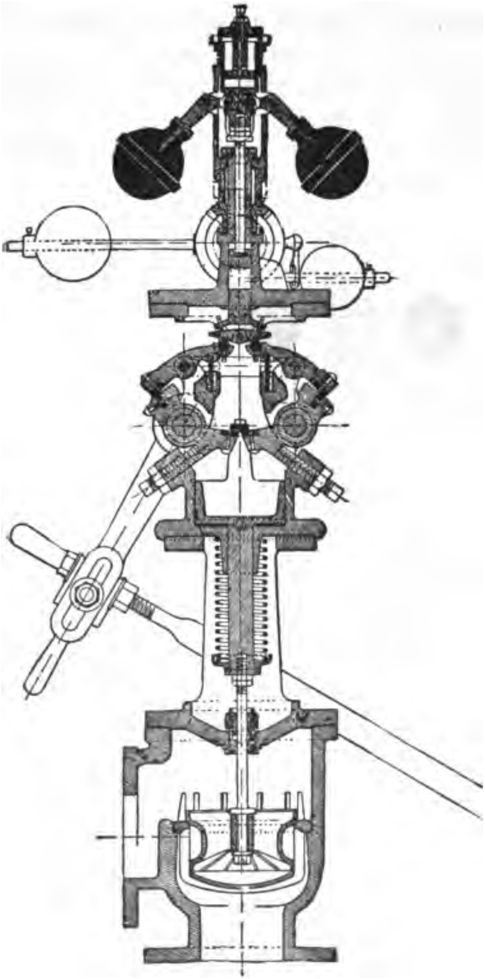
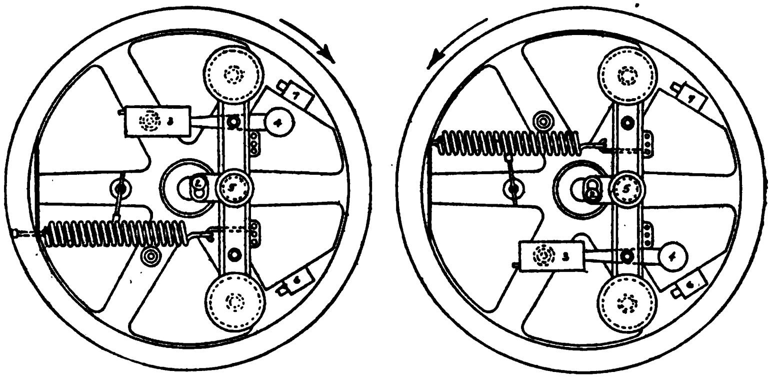


FIG. 708.—Nordberg automatic cut off governor. This device is a combination of trip *cut off gear* and a *governor* for controlling the same, designed to be attached to the steam pipe of slide valve, rocking valve and similar engines to regulate the speed of the engine. The steam is admitted at full boiler pressure; is cut off at a point corresponding to the demand for power, and expanded. ***In construction,*** the governor consists of a double beat poppet valve, operated by a double trip mechanism. A sensitive regulator sets the point of cut off, according to the demand for steam. The range of cut off obtainable is from 0 to ¾ or ⅞ of stroke. An air dash pot causes the valve to drop gently on the seat. All contact edges about the trip gear are made in shape of removable hardened plates, of best English steel. These plates are reversible, and all eight edges can be used as contact edges. The cut off gear is operated by an independent eccentric furnished with the machine; the regulator is driven by a belt. A safety stop is provided which will stop the engine in case of any accident to the governor belt, or if the regulator should stick. The safety stop will keep a uniform tension on the governor belt. Speed of governor up to 200 *r.p.m.*

The conditions of load vary widely in different classes of work. In the case of factory or mill engines, the load is practically constant, while with those employed for electric railway work, the load changes constantly as the various cars along the line are started and stopped. In the case of rolling mill engines, the conditions are even worse, for here the engine may be running light, with no load except its own friction, when suddenly the rolls are started and the maximum load will be thrown on at once.

With any governor of whatever type, there must be a certain variation in the speed of the engine to operate it. In most well designed engines the speed will not vary more than two per cent above or below the mean speed, and in many cases even closer regulation is obtainable.



FIGS. 709 and 710.—*McEwen* right hand engine governor; fig. 363 run *over* setting; fig. 364, run *under* setting. The wrist pin is shown at 2; there is another hole in the arm directly below. 3 is a dash pot, and as centrifugal force interferes with its free operation at high speeds the weight 4, is provided to counterbalance it and prevent undue friction. The governor arm is pivoted at 5 and rubber buffers are provided at 6 and 7, the ends of travel. Fig. 364 shows gear reversed. The wrist pin is in the other hole provided for it

A regulation within 4 per cent means that if the normal speed of the engine is 100 *r.p.m.* at its rated capacity, it should not rise above 102 *r.p.m.* when all the load is thrown off, nor drop below 98 *r.p.m.* when the maximum load for which it is designed is thrown on.

Ques. What is close regulation?

Ans. Small speed variation.

Ques. What is the usual regulation in practice?

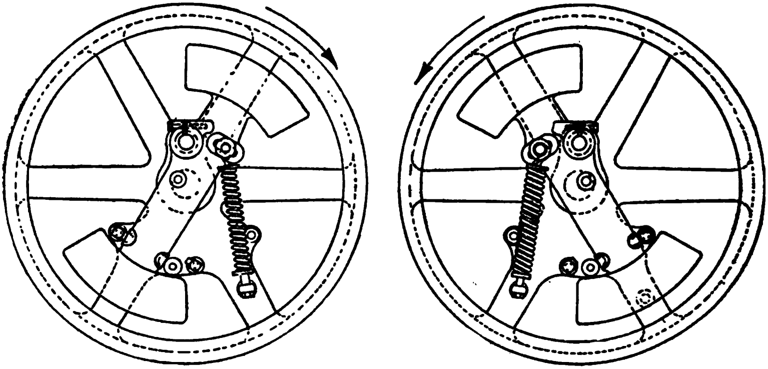
Ans. From 2 to 4 per cent.

Shaft Governors.—This type of governor is used chiefly in that large class of engines popularly known as automatic cut off engines. Because of the high rotative speed, a powerful and sensitive governor can be provided without undue weight or vibration.

Shaft governors may be classified:

1. With respect to the controlling force or forces, as

   a. Centrifugal;
   b. Inertia;



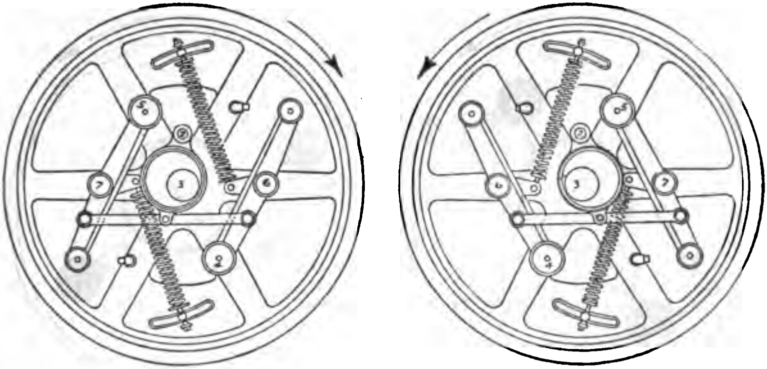
FIGS. 711 and 712.—*Rites* governor as applied to Watertown engine. Fig. 365, governor in forward motion; fig. 366 governor reversed.

   c. Combined centrifugal and inertia;
   d. Combined inertia, and neutralized centrifugal.

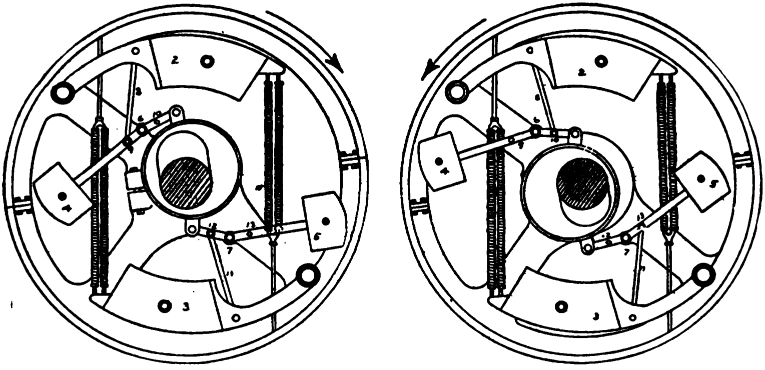
2. With respect to the regulating mechanism, as

   1. Variable throw;
   2. Variable angular advance;
   3. Combined variable throw and variable angular advance.

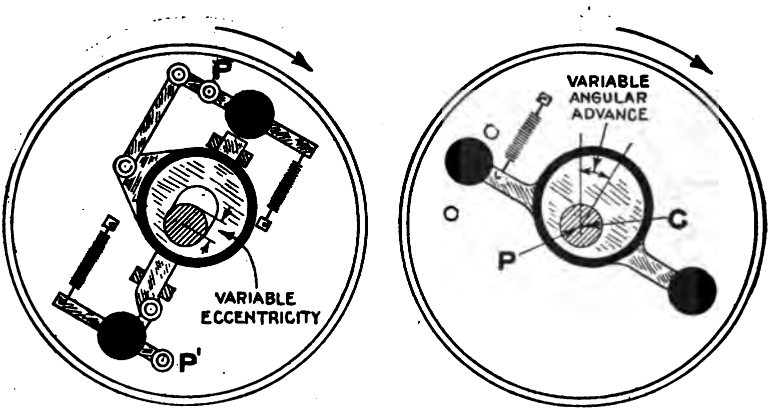
Many of the so called inertia governors are, in fact, of the



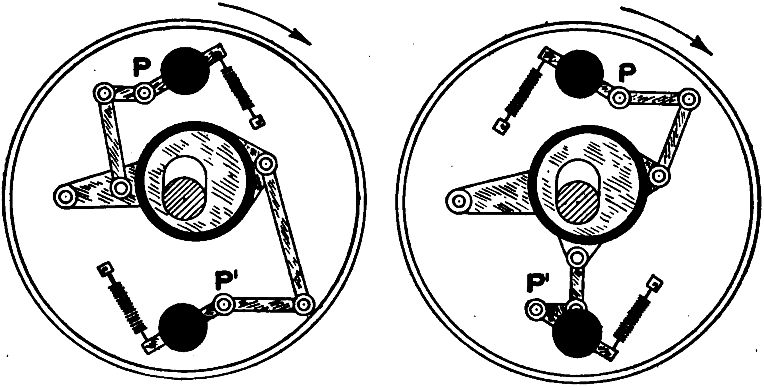
FIGS. 713 and 714.—***Harrisburg*** governor in forward and reversed motion. The eccentric is pivoted at 2, and swings across the shaft 3, by action of the weights 4 and 5, which are pivoted at 6 and 7, respectively. After reversing governor as in fig. 714, if sluggish, move the ends of the spring toward the rim of wheel; if super-sensitive, move them toward the center.



FIGS. 715 and 716.—***Fitchburg*** governor in forward and reverse motion. When in the position shown the governor is at rest, and the springs draw the heavy weights 2 and 3, inward, thus raising the eccentric to its maximum eccentricity. ***In operation,*** *centrifugal force* throws the weights 2 and 3, outward against the force exerted by the springs, reducing the eccentricity until equilibrium between cut off and load is established. The auxiliary weight 4, operates with the main weight 2, as it is pivoted at 6, but the other auxiliary weight 5, operates against the main weight 3, as it is pivoted at 7, hence 4 and 5, balance each other, but taken together they resist changes in speed, therefore the result is a very steady speed under variable load. If in fig. 715, link 8, be disconnected from pin 9, and connected at 10, the eccentric will be moved in the opposite direction by centrifugal force. By disconnecting link 11, from 12, and connecting it at 13, it also reverses motion because it is pivoted at 7, these changes reversing the governor.



FIGS. 717 and 718.—Shaft governors illustrating principles. Fig. 717, centrifugal control, variable throw regulation; fig. 718, inertia control, variable angular advance regulation.



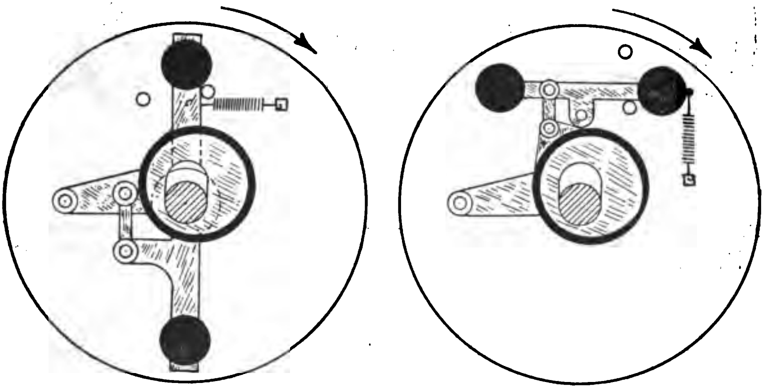
FIGS. 719 and 720.—Shaft governors illustrating principles. Fig. 719, combined centrifugal and inertia control (forces acting together), combined variable throw and variable angular advance regulation; fig. 720, combined centrifugal and inertia control (forces acting in opposition), combined variable throw and variable angular advance regulation.

combined centrifugal and inertia class, these forces acting either in unison or in opposition according to construction.

The principles relating to the controlling forces have been explained under inertia governors, and the methods of varying the cut off by altering the travel or angular advance, have been treated at length in chapter 6 on variable cut off.

Figs. 717 to 720 show four types of shaft governor illustrating the controlling forces and regulating mechanism as classified.

In fig. 717 the action depends on centrifugal force alone. Inertia acts along the axis through the pivots P P′, and therefore does not tend to rotate the ball around the pivot. The regulating mechanism changes the cut off by variable throw.



Figs. 721 and 722.—Shaft governors illustrating principles. Fig. 721, inertia control, variable throw and variable angular advance regulation; fig. 722, combined inertia and neutralized centrifugal control, variable travel and variable angular advance regulation.

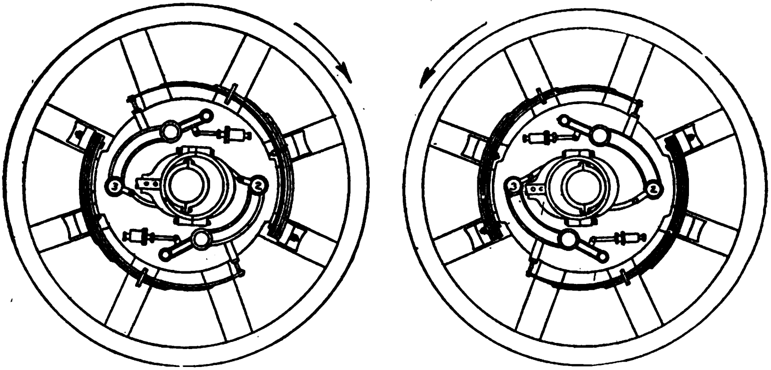
The governor illustrated in fig. 718, controls the engine speed by inertia alone, the two forces acting together; regulation being by variable angular advance.

Figs. 721 and 722 show a regulating mechanism with swinging eccentric which regulates by combined variable angular advance and variable throw. The control is by combined centrifugal force and inertia, these forces acting together in fig. 721, and in opposition in fig. 722.

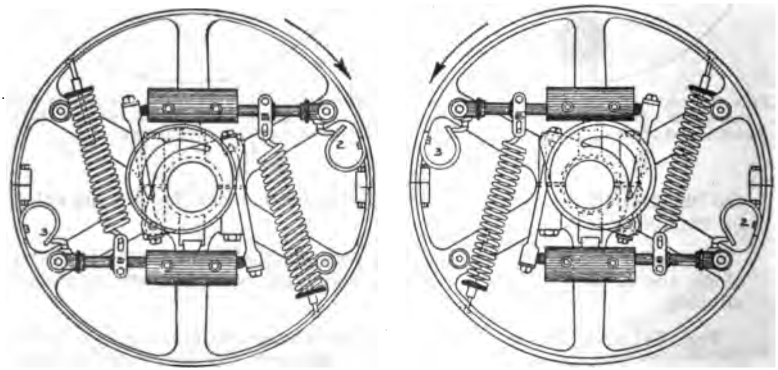
Fig. 721 shows a control by inertia alone, the regulation being by variable angular advance.

Fig. 722 is an example of combined inertia and neutralized centrifugal control with variable angular advance regulation.

It should be understood that the series of illustrations, figs. 717 to 722, are intended to represent principles rather than construction, and are therefore to be regarded as elementary diagrams.

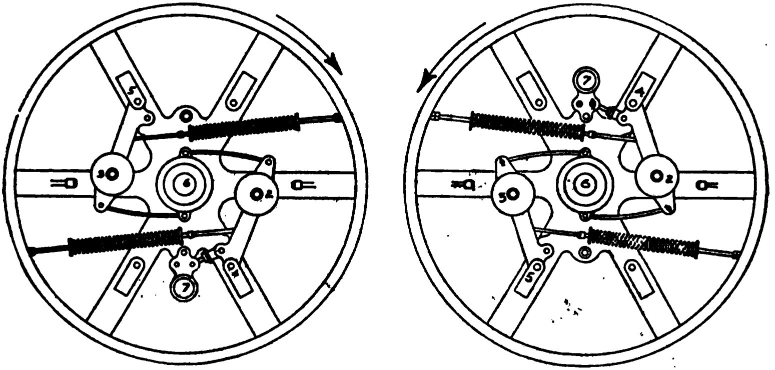


FIGS. 723 and 724.—***McIntosh and Seymour*** governor in forward and reverse motion. ***In operation,*** centrifugal force throws the weights 2 and 3, outward giving *variable angular advance.*

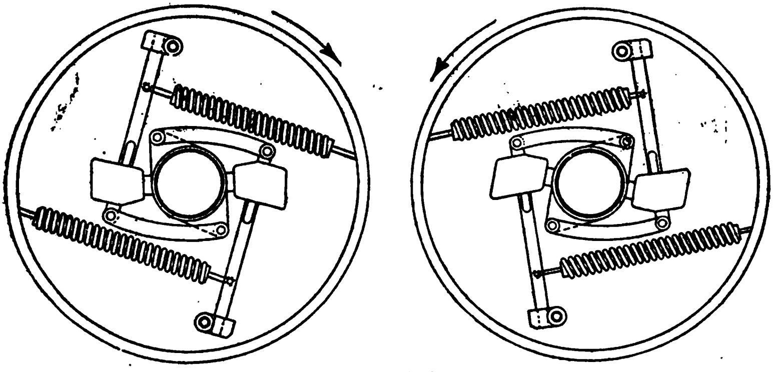


FIGS. 725 and 726.—***Buckeye*** governor in forward and reverse motion. *Centrifugal* control and *variable angular advance* regulation. ***To reverse,*** all connections are put in opposite places.

Auxiliary Devices.—Some governors are fitted with dash pots or other damping devices to prevent super-sensitiveness. All governors should be provided with a safety stop, or device



FIGS. 727 and 728.—***Clark** combined centrifugal and inertia* governor in forward and reverse motion. ***Regulation** is by variable angular advance.* The centrifugal weights 2 and 3 are pivoted at 4 and 5 respectively. *Inertia* control is secured by the weight 7 pivoted radially at 6.



FIGS. 729 and 730.—***Russell** centrifugal* governor in forward and reverse motion. ***Regulation** is by variable angular advance.* To reverse this engine the main eccentric must be turned, and if an offset key be used it must be reversed. The cut off eccentric must be carried around on the shaft until it has the same angular position in advance of the crank that it had before. It is necessary to reverse the spring connections and weighted arms to accomplish this. The eccentrics are made in halves to facilitate their removal.

which closes the throttle in case the belt or drive gear should break. A desirable feature is a speed regulator which permits wide adjustment of the speed during operation.

These devices are illustrated in the various cuts of governors as constructed by the various manufacturers.

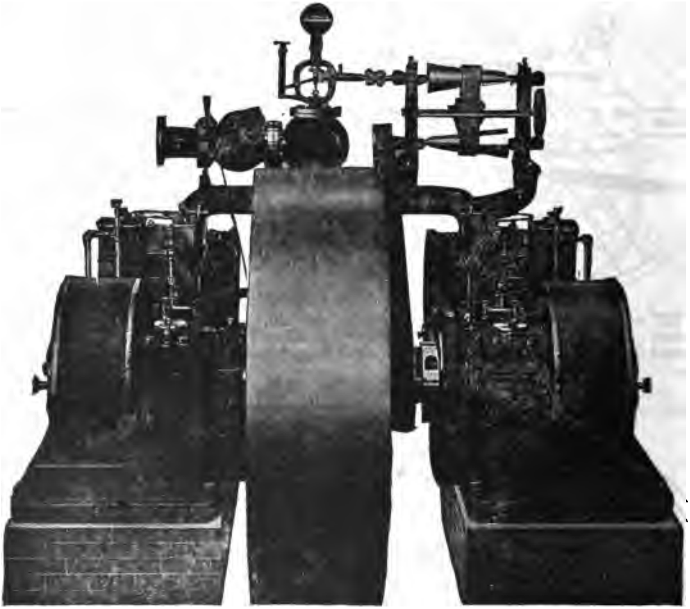


Fig. 731.—Variable speed changing cones as applied to the Ball variable speed engine. ***In operation,*** the speed of the upper cone is constant while that of the lower (together with the engine speed), varies as the cone belt is shifted to the right or left by means of the shifter operated by the hand wheel.

Variable Speed Governors.—There are some conditions of service, as in paper mills, where it is frequently necessary to vary the speed of the engine within a wide range. Engines fitted with governors designed especially for speed variations are known as variable speed engines, and are intended for all classes of manufacturing where the quality, thickness or weight of the manufactured product is affected by the speed at which the machinery runs. The governor is usually of the throttle type fitted with a pair of variable speed cones as shown in fig. 731, or with friction discs as in fig. 732.

Since the speed of the governor must always be constant, no matter what speed is required of the engine, evidently the speed changing part of the mechanism is some form of transmission between the engine shaft and the governor by means of which the ratio of gearing may be varied, just as, for instance, is done in the transmission or gear box of an automobile, only in the case of the variable speed governor, there must be possible an infinite number of gradations, or speed changes, instead of three or four as in the automobile.

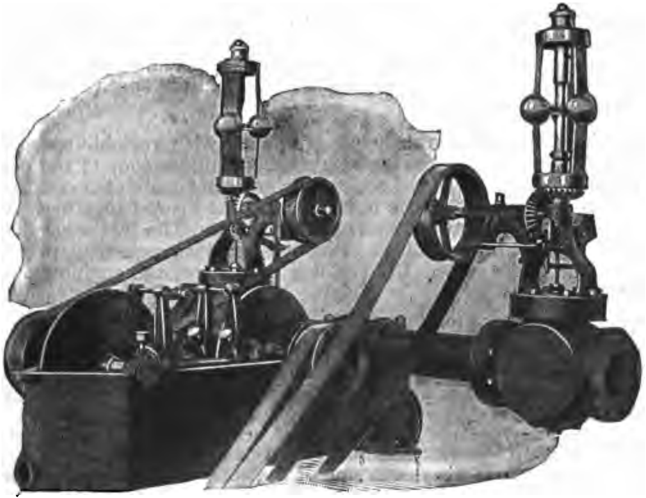


Fig. 732.—American Ball variable speed mechanism and automatic stop as applied to paper mill engine. At the left in the rear is the speed governor driver through variable speed friction discs, and at the right, the automatic engine stop driven directly from the engine. The steam valve of the automatic stop remains wide open throughout the whole normal range of speeds for which the engine is designed, but in case the speed exceed the predetermined limit, the mechanism is tripped and the weighted lever closes the steam valve. At all speeds within range of the variable speed device, the automatic stop has no throttling effect and hence cannot affect the speed of the engine.

Referring to the principle just stated, there is a speed stamped on every throttling governor and the valve of the governor will not close until that speed is reached. Different sizes and styles of governors are stamped for different speeds, but the same principle applies to all.

To illustrate this, if the governor of an ordinary throttling engine were stamped at 200 revolutions, and it was required to run the engine at 100 revolutions, it would be necessary to put a governor belt pulley on the governor shaft one-half the diameter of the governor belt pulley on the engine shaft, that is, the diameter of the two pulleys would have to be such that, when the engine was running at 100 revolutions the governor would be running at 200 revolutions; because, as stated, the governor will not regulate the steam supply until it has reached the speed which the manufacturers stamp upon it.

In showing how this principle is applied with variable speed cones, as in fig. 731, it must be remembered that the upper cone of the pair always runs at a constant speed because the governor runs at a constant speed; the governor shaft and the upper cone being geared together. With this in mind, suppose that the short belt connecting the two cones is so placed that it is at the big end of the lower cone, and consequently, is at the small end of the upper cone. Under these conditions, the engine would run at a very slow speed.

The reverse of this situation would occur when the short belt between the cones is shifted to the small end of the lower cone, and consequently is at the large end of the upper cone. Under these conditions, the engine would run very fast, in order to keep the speed of the governor at the speed stamped upon it.

This belt, which connects the two cones, is held in a frame, and this frame is moved from one extreme of the cone to the other, by means of a long screw turned at the end by a hand wheel. This hand wheel is shown in the figure at the right of the open engine. By turning this wheel, the belt is gradually moved along the cones, from one end to the other, and this movement causes the engine to run at varying speeds in order that the governor may always run at the constant speed stamped on it.

By means of this device it will be seen that the speed of the engine is regulated with the utmost nicety, increasing or decreasing gradually and without the slightest shock to the driven machinery, and without shutting the engine down. Since the cones are tapered instead of being stepped, the number of possible changes between the two extremes is without limit.

If it be desired to regulate the speed of the engine from some other part of the room, the hand wheel, which operates the screw of the regulating device, may be replaced by a sprocket wheel and chain.

Usually a graduated scale and pointer are provided on the speed changing device, so that the engine may be set to run at any desired speed, or changed from one speed to another. without using a speedometer.

Since a variable speed engine usually operates machinery worth many times the value of the engine, a *secondary speed control* is provided, which provides additional means of preventing damage to the driving machinery in case of accident.

One type of secondary speed control consists of a quick closing emergency stop valve in the steam line, which is automatically closed by a tripping device, attached to the rim of the band wheel. This tripping device flies out under high speeds and releases a catch which is connected by a rod to the weight lever of the emergency stop valve. When released this weight on the lever of the valve swings to its lowest position and shuts off the steam.

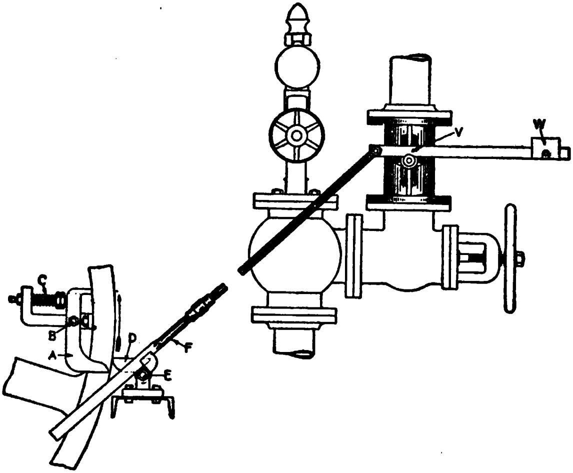


Fig. 733.—Chandler and Taylor trigger device for *secondary speed control.* ***In operation,*** when the speed of the engine exceeds a predetermined rate, centrifugal force causes the lever A, pivoted at B, to overcome the compression of the spring C, and fly out engaging the trip D, pivoted at E, which releases the steam arm F, allowing the weight W to fall and close the butterfly valve V.

A second type of secondary speed control is the shaft governor connected to a swinging eccentric and so developed that it will not become operative until the speed of the engine arrives at the maximum speed at which it is to run. Any effort of the engine to run beyond this maximum speed shortens the valve travel and by so doing cuts off the steam in the steam chest. In this device the engine is not brought to a stop as with the trigger device. In this second plan the engine operates at its maximum speed, until it is shut down by hand.

The objection to the trigger device is not in the device itself, but in the neglect of it. If the trigger in the wheel be not kept clean and free it is apt to gum up and stick, and thus fail to perform its functions, when called on to do so. The only objection tc the shaft governor device is its increased expense. There are no pins to stick in the shaft governor, and the weights in the wheels, rotating at the high speeds which they do, have sufficient power to take care of any emergency.

In the operation of paper machines the shaft governor has still another advantage: When the engine speeds up to its maximum speed, the operator can tear off the line of paper at a point before it enters the drying rolls, and so allow the paper in the dryers to run itself out and clear the machine before engine is shut down.

With the trigger device the engine is shut down at once, leaving all the paper in the machine.

Governor Troubles.—A governor with its delicate control and regulating mechanism is a delicate piece of apparatus and must be kept in first class condition to secure satisfactory working.

Dry pins often give trouble by introducing excess friction which opposes the controlling force. As much alteration should be given to governor lubrication as to any other part of the engine. Special attention should be given to the lubrication of eccentrics, and the main pin on inertia governors.

Lost motion in the various joints is not conducive to best operation.

The engineer should frequently move the governor weight arms by hand when the engine is not running; any undue force required in this operation on small and medium size engines will indicate that some part of the apparatus is not in proper working order.

Every time the valve stem packing is tightened more resistance is added which must be overcome by the controlling force, hence for satisfactory governor operation the valve stem packing should be kept in as near one adjustment as possible.

The valve setting or its condition is quite as important as other causes of trouble. See if the valve be set properly or if it leak, or the pressure plate bind.

If the normal load be such that the valve does not over travel the seat, a shoulder will be worn near the seat limit, hence a slight increase of load causing more travel will cause the valve to ride on the shoulder and leak. The valve seat should be carefully examined for this defect.

When springs are too weak to give the required tension, cut off enough turns to produce the proper effect.

CHAPTER 12

PUMP VALVE GEARS

Principles of Operation.—Many ingenious valve gears have been devised for operating the steam ends of direct acting pumps. The need for such special devices is caused by the absence of a rotating part which prevents the use of an eccentric. In most cases, the necessary movements of the valve gear are obtained from two sources:

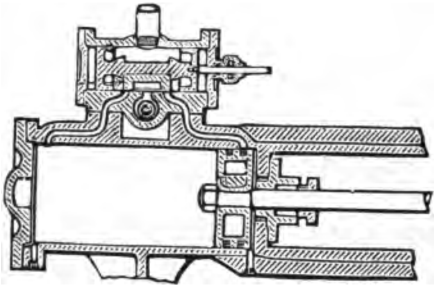
1. The movement of the piston or piston rod;
2. The steam pressure.

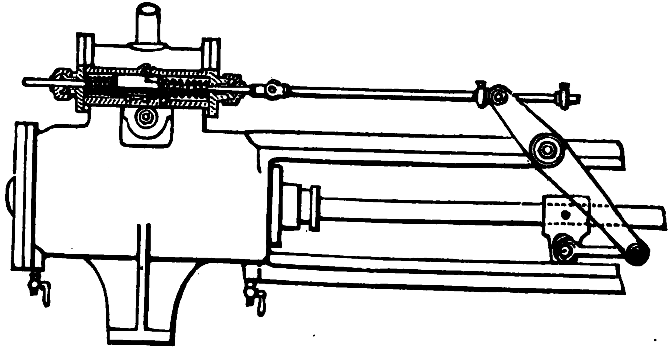
The valve gear when thus operated usually consists of:

1. A main slide valve which admits and exhausts steam from the cylinder;
2. An auxiliary piston connected to the main valve and moving in a cylinder formed in the valve chest;
3. An auxiliary valve controlling the steam distribution to the auxiliary piston cylinder, and operated with suitable gear by the main piston, or piston rod.

The cycle of operation of a valve gear of this type is as follows:

As the main piston approaches the end of the stroke, it moves the auxiliary valve which causes steam to be admitted to one end of the auxiliary piston and exhausted from the other. This results in a movement of the auxiliary piston which in turn

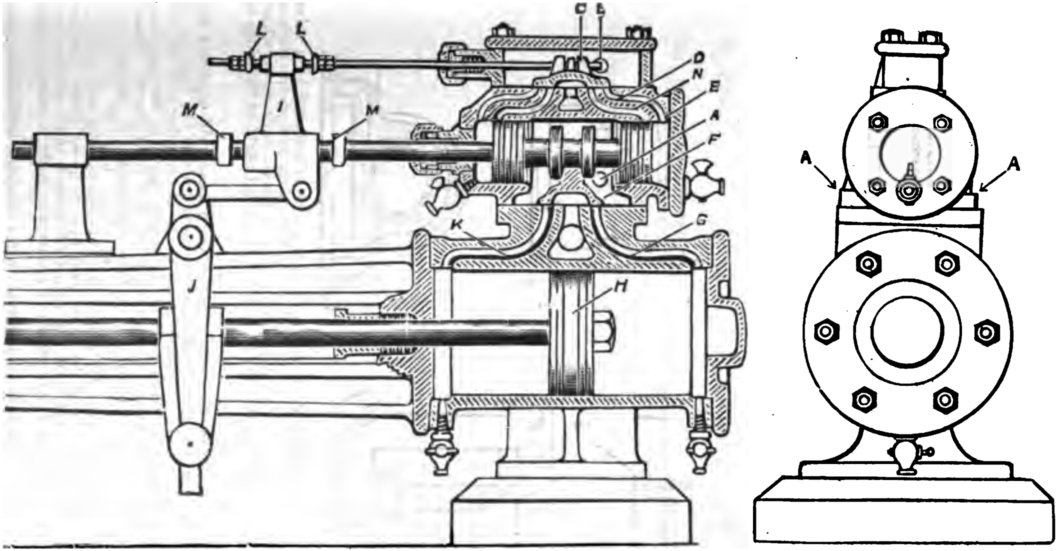




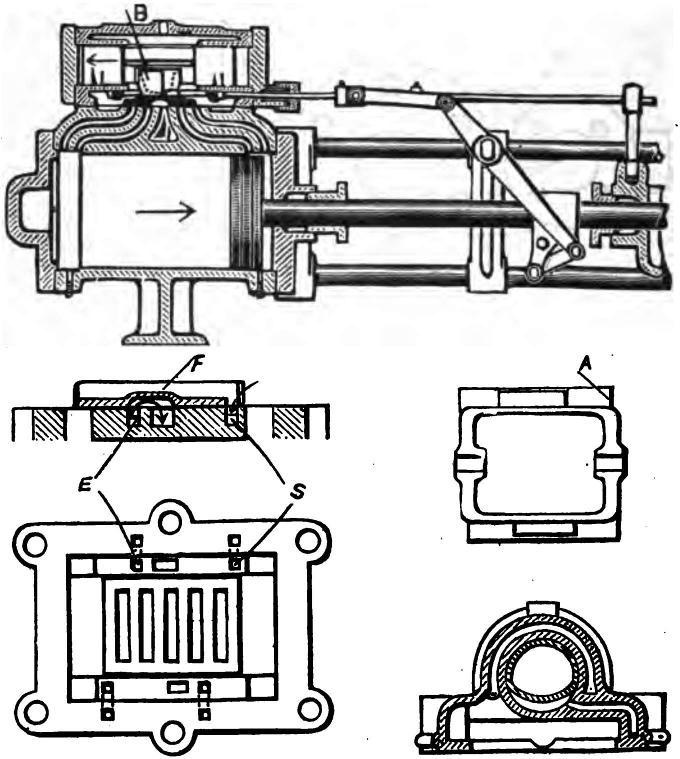
FIGS. 734 and 735.—Valve gear of the *Snow* pump. The auxiliary valve is a plain flat slide operated by a valve stem, the latter being moved back and forth by means of a rocker shaft, as shown, the upper end of which alternately comes in contact with the collars on the stem. The outer end of the valve stem passes through a sleeve attached to a pin in the upper end of the rocker arm, as shown. A knuckle joint near the stuffing box permits the rod to vibrate without causing any derangement in the alignment of valve stem through the stuffing boxes. On the valve stem at either end of the auxiliary valve is a spring, which tends to keep the valve in a central position, so that when the rocker arm engages one of the collars, the valve is drawn against the spring toward that end of the stroke. The result is that the stem and valve follow the rocker arm on the return stroke to its mid-position, and are started on the latter half of the stroke by the stem, but without shock or lost motion. This arrangement is particularly valuable in the case of condensers, and in pumps where the first part of the stroke is made quickly, and the piston is then suddenly stopped by coming in contact with a solid body of water, the latter part of the stroke being made much more slowly. The springs on either side of the auxiliary valve take up lost motion and keep the parts in contact, thus preventing shocks and unnecessary wear.

The auxiliary valve controls the admission and exhaust of steam from the steam chest and valve piston in the manner common to all slide valve engines. The valve piston is connected to the main valve, which allows the valve to find its own bearings on the seat and not only takes up the wear automatically, but produces even wear.

To set the auxiliary valve, see that the valve is in its central position when the rocker arm is plumb, and that the collars on the valve stem are located at equal distances from each end of the sleeve. When the piston moves to one end of the stroke, the auxiliary valve will open the small port at the opposite end, provided the collars on the valve stem have been properly placed. Setting the collars closer together shortens the stroke of the piston, and moving them farther apart lenghens the stroke. The piston should always make a full stroke without danger of striking the cylinder heads.



Figs. 736 and 737.—Valve gear of the *Buffalo* pump. As shown, the piston and valves are in their central position, which condition never occurs in operation; if it did, the pump would stop. The valves and pistons begin at one end or the other of the stroke uncovers the ports, and the moment steam is admitted the pump will start. In the figures A, is the main steam pipe, and B, the auxiliary steam pipe. These pipes are one, inside the casting, so that one pipe supplies both. Assume the valve, C, moved to the left so that the port D, is uncovered. Live steam then flows through the port D, and pushes the balanced piston valve E, to the left, carrying the slide valve F, with it to the left, so that the port G, is opened to the steam cylinder. The steam enters through the port G, pushes the main piston H, to the left, completing one stroke of the piston. As the piston travels to the left the lever J, pushes the cross head I, to the right, which opens corresponding ports in the left hand end of the cylinder, and the piston valve E, is pushed to the right, which admits steam on the left hand side of the main piston through the port K, pushing the piston to the right, completing the second stroke of the piston. The auxiliary slide valve C, is operated by the cross head I, coming in contact with the tappets L L. The auxiliary tappets and steam M M, are provided because the valve E, might stick, due to the pump standing for some time unused or from some other cause. In such contingency the cross head I, is pushed to the right by the action of the main piston and comes in contact with the



FIGS. 738 to 742.—Valve gear of the ***Deane*** pump. The main valve is operated by a valve piston without lost motion, as in fig. 738. Steam is admitted alternately to opposite ends of the valve piston. The motion of the piston valve is controlled by a secondary valve, which admits and exhausts the steam to the valve piston through the small ports at the sides of

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FIGS. 736 and 737—*Continued.*

tappets, M M, which causes the piston valve to start, after which steam will complete the work. When the pump is running, the cross head I, never quite touches the tappets, M M, because it engages the tappets L L, admitting steam to the piston valve and shifts it before the tappets, M M, are touched. The reason of the double ports in the auxiliary steam chest is to have one port D, for steam, and one port N, for the exhaust. Steam being imprisoned between these two ports forms a cushion, preventing the piston valve striking the heads of the chest. The tappets L L, set closer together or farther apart control the stroke of the main piston H. When the pump is running very fast, the momentum of the moving parts increases and the tappets will have to be set closer together for high speed than for slow. The tappets M M, are adjustable to their right relation with the tappets L L. The general design and easy means of adjustment make a reliable single cylinder valve motion.

To set the valves.—There are no complicated internal parts requiring adjustment, and almost all parts requiring manipulation can be handled while the pump is running.

moves the main valve. The steam distribution to the main cylinder thus affected, reverses the motion of the main piston, and the return stroke takes place, completing the cycle.

In pumps of different makes, the detail which varies mostly is the auxiliary valve and the method by which it is operated. With respect to these features the majority of pumps may be divided into two classes, as those having:

1. A separate auxiliary valve;

2. Auxiliary valve and auxiliary piston combined.

In pumps of the first mentioned type, the auxiliary valves usually have stems or tappets which project into the cylinder at the ends and are moved by contact with the main piston as it nears the end of the stroke.

FIGS. 738 to 742—*Continued.*
the steam chest. The secondary valve derives its motion from the valve stem, tappets, links and the piston rod as shown.

In operation, assume the piston moving in the direction of the arrow near the end of the stroke; the tappet block comes in contact with the left hand tappet and throws the secondary valve to the left until its edge, A, fig. 741, uncovers the small port, S, figs. 739 and 740, admitting steam to the valve piston. The port, E, and chamber, F, in the secondary valve provide for the exhaust of steam from the left hand end of valve piston in the same manner and at the same time that steam is admitted behind the right hand end. The exhaust ports in the chest allow for properly cushioning the valve piston. The small ports on the other side of the steam cylinder, figs. 739 and 740, control the motion of the valve in the other direction and act in exactly the same manner as those just described. In case the steam pressure should fail to start the valve piston in time, a lug, B, fig. 738, which forms a part of the valve stem, comes in contact with the valve piston and the entire power of the steam cylinder starts it. The correct timing of the valve movements is controlled by the position of the tappets. If they be too near together, the valve will be thrown too soon and thus the stroke of the pump will be shortened, while on the other hand, if too far apart, the pump will complete its stroke without moving the valves. These tappets are set and keyed securely before leaving the factory The exhaust from the cylinder is cut off when the piston covers the inner port, and forms a steam cushion for the piston to prevent it striking the heads.

To set the valve —Place the steam piston at the end of stroke nearest stuffing box and the secondary valve so that it will uncover the steam port, S, figs. 739 and 740. Set the tappet next to the steam cylinder on the valve stem against the tappet block and secure it in this position. Slide the secondary valve forward until the opposite steam port is uncovered and place the steam piston in its extreme outward position, then set the other tappet against the tappet block. Now set the valve so that the inside main steam port is open and the valve piston in position to engage the main steam valve, put the valve chest on the cylinder and secure it in place. The pump will then be ready to start on the admission of steam to the steam chest. If when steam is turned on, the pump refuse to start, simply move the valve rod by hand to the end of its stroke and the pump will move without trouble. In renewing the packing between the steam chest and cylinder, caution should be observed to cut out openings for the small ports.

Where the auxiliary valve is combined with auxiliary piston, an initial rotary motion is given the latter by the external gear, causing it to uncover ports which give the proper steam distribution for its linear movement.

An example of the separate auxiliary valve type is shown in fig. 743 which illustrates the steam end of the Cameron pump.

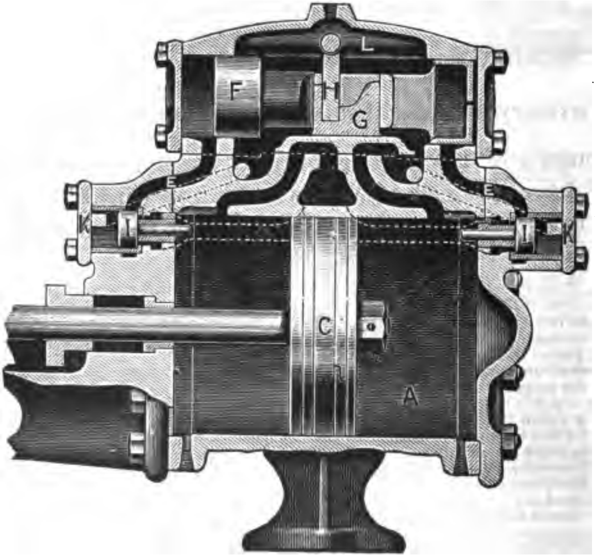
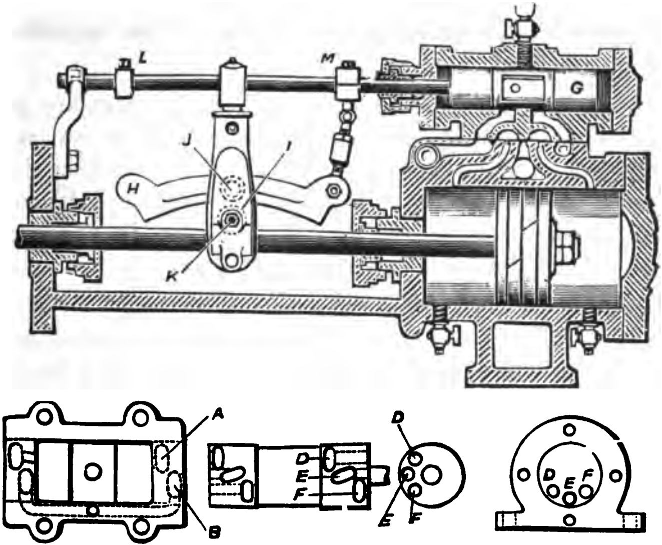


FIG. 743.—Valve gear of the Cameron pump; an example of the *separate auxiliary valve* class. ***Its construction*** and operation is explained in detail in the accompanying text. **No valve setting is necessary,** it being only necessary to keep the valves I, I, tight by occasional grinding. ***In operation,*** the piston as it nears the end of each stroke strikes the stem and lifts the valve off its seat; this allows the exhaust steam behind the piston valve to escape. The live steam pushes the piston towards the exhausted end carrying the main slide valve along with it.

Each auxiliary valve I, has a short stem which projects into the cylinder; when the piston C, strikes one of these, the valve is driven back, and opens an exhaust passage E, from the corresponding end of the auxiliary piston F, which immediately



FIGS. 744 to 748.—Valve gear of the ***Knowles*** pump. The valve piston is driven alternately backward and forward by the pressure of steam, carrying with it the main valve, which admits steam to the main steam piston that operates the pump. The main valve is a plain slide whose section is of B form, working on a flat seat. The valve piston is slightly rotated back and forth by the rocker bar H; this rotative movement places the small steam ports D, E, F, figs. 746 to 748, which are located in the under side of the valve piston in proper position with reference to the corresponding ports A, B, cut in the steam chest. Steam enters through the port at one end and fills the space between the valve piston and the head, drives the valve piston to the end of its stroke and carries the main slide valve with it. When the valve piston has traveled a certain distance, a corresponding port in the opposite end is uncovered and steam enters, stopping its progress by giving it the necessary cushion. There is no dead center.

In operation, the piston rod with its tapped arm J, fig. 744, moves backward and forward with the piston. At the lower part of this tappet arm is attached a stud or bolt K, on which is a friction roller I. This friction roller, lowered or raised, adjusts the pump for a longer or shorter stroke. This roller coming in contact with the rocker bar at the end of each stroke, and this motion is transmitted to the valve stem, causing the valve to roll slightly. This action opens the ports, admits steam and moves the valve piston, which carries with it the main slide valve which admits steam to the main piston. The upper end of the tappet arm does not come in contact with the tappets L, M, on the valve rod, unless the steam pressure from any cause should fail to move the valve piston, in which case the tappet arm moves it mechanically.

To set the valve, loosen the set screws in the tappets on the valve stem. Then place the piston at mid-stroke, and have the rocker bar H, in a horizontal position, as shown in the engraving. The valve piston should then occupy the position shown in fig. 748. The valve piston may be rotated slightly in order to obtain this position by adjusting the length of connection between the rocker bar H, and the valve stem. Then turn the valve piston G, one way or the other to its extreme position, put on the chest cover, and start the pump slowly. If the pump make a longer stroke on one end than on the other, lengthen or shorten the rocker connection so that the rocker bar H, will touch the rocker roller I, equally distant from the center pin J. If the pump hesitate on the return stroke, it is because the rocker is shifted under the action of live steam on the opposite side of the piston head.

There is a small hole in each end of the auxiliary piston, and when both auxiliary valves are closed, the steam passing through these holes leaves the auxiliary piston entirely surrounded by live steam, and therefore in perfect balance endwise, until the main piston strikes the stem in the opposite cylinder head, when the valve moving operations are repeated in the opposite direction.

The space back of the auxiliary valve communicates with the steam chest, through a passage shown in dotted lines; the valve is therefore closed by steam pressure as soon as the piston moves back from the stem.

It should be noted that the piston closes the exhaust passage before the end of the stroke. The confined steam forms a cushion between the piston and the cylinder head, but a little passage is cut in the cylinder wall through which sufficient steam is admitted to start the piston on the return stroke.

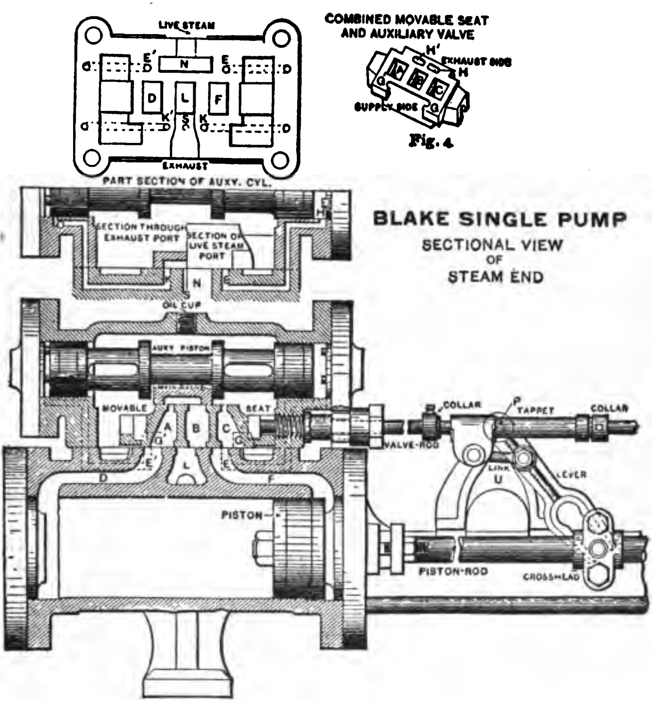
The auxiliary piston which carries the main valve G, shifts the latter in the direction the piston travels at the end of the stroke, that is, opposite to that of a common slide valve. This valve has, therefore, two cavities, each of which alternately puts the cylinder in communication with the steam chest and the central exhaust port. Steam is admitted under the outer valve face, as shown in the figure.

H, is a lever, by means of which the auxiliary piston may be reversed by hand when expedient.

An example of the second mentioned type in which the auxiliary valve and auxiliary piston are combined is shown in figs. 753 to 755, illustrating the valve gear of the Davidson pump.

FIGS. 744 to 748.—*Continued.*

roller, I, is too low and does not come in contact with the rocker bar, H, soon enough. To raise it, take out the rocker roller stud, K, give the set screw in this stud a sufficient downward turn, and the stud with its roller may at once be raised to its proper height. If the valve rod tremble, slightly tighten the valve rod stuffing box nut. When the valve motion is properly adjusted the vertical arm should not quite touch the collar L, and the clamp M. Rocker roller I, coming in contact with the rocker bar H, reverses the stroke.



Figs. 749 to 752.—Valve gear of the Blake pump. The main valve, is carried by the auxiliary piston, and moves on the back of the movable seat (fig. 752), the passages A B C, of which serve as steam passages. The lugs G G′, control admission of steam to the auxiliary cylinder, and the holes H H′, control the exhaust from that cylinder. ***In operation,*** when the piston nearly reaches the left end of the cylinder, the movable seat is shifted to the left so that lug G, covers the port E, while lug G′, uncovers port E′, thus admitting steam behind the auxiliary piston at the left side. At the same time the exhaust port K, of the auxiliary cylinder is opened to the hole S, leading to the exhaust, and forcing the auxiliary piston over to the right, uncovering port A, to live steam. Near completion of the stroke the operation is reversed. The auxiliary piston is cushioned on steam, because the exhaust port is not out at the end of the auxiliary cylinder, and consequently there is steam imprisoned when the piston covers the exhaust, as at the left in fig. 751. The main piston is cushioned on live steam, because the valve has lead; that is the operation of admitting steam is performed before the piston reaches the end of its stroke. It will be seen that if means be provided to shift the movable seat from one end of its travel to the other, the rest of the operation is automatic. Fig. 752, shows the valve gear provided for this operation. The piston rod is provided with a cross head, the latter having a pin as shown. The frame of the pump is built with an upright piece U, to which is pivoted at P, a lever whose lower end is slotted and engages with the cross head. The valve rod, which is secured to the movable seat, is provided with two collars as shown. These collars are made of split nuts which work on a thread cut on the valve rod for a short distance on each side of their ordinary position. Between these two collars is a tappet, which is free to slide on the valve rod. The link shown connects the tappet with the

The main valve is operated by a positive mechanical connection between it and the main piston rod, also by the action of steam on the valve pistons. Fig. 753, shows the details of valve gear and steam cylinder. In the figures the steam end consists of the cylinder M, valve A, and valve pistons B and B. These pistons are connected with sufficient space between them for the valve A, covering the steam ports F and F_1, as in fig. 755.

The valve is operated by the steel cam C, acting on a steel pin D, which passes through the valve into the exhaust port N, in which the cam is located. In addition to this positive motion, steam is alternately admitted to and exhausted from the ends of the valve piston through the ports E and E_1, which moves the pistons B and B_1.

Assuming the pump to be at rest with the valve A, covering the main steam ports F and F_1, in which position the cam C, holds the main valve by means of the valve pin D, so that ports E and E_1, admit steam to one end of the valve piston at the same time connects the other end with the exhaust port. The steam, acting on the valve pistons, moves both, opening the main ports F and F_1, admitting steam to one end of the steam cylinder and opening the other end to the exhaust.

If the valve occupy any other position than the one described, the main ports, F and F_1, will be opened for the admission and exhaust of steam; consequently it is evident that this pump will start from all points of the stroke.

On the admission of steam to the cylinder the main port F, the main piston, cam and valve will move in the direction indicated by the arrows. The first movement of the cam oscillates the valve, preparatory to bringing it into a proper position for the opening of the auxiliary steam ports E, to live steam, and E, to exhaust, also to close the valve mechanically just before the main piston reaches the end of its stroke. This causes a slight

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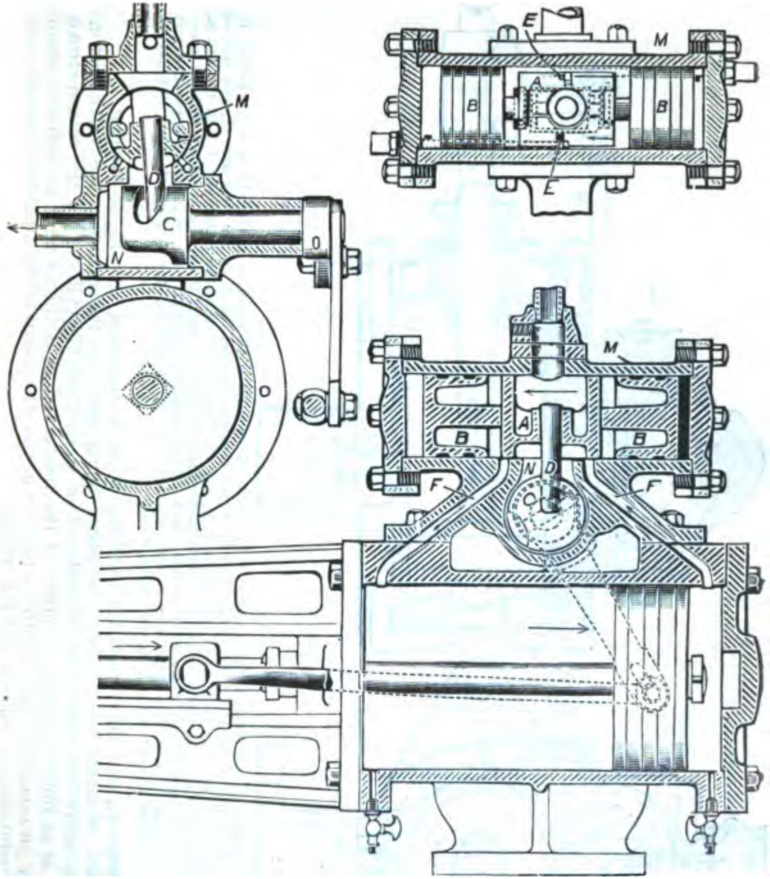
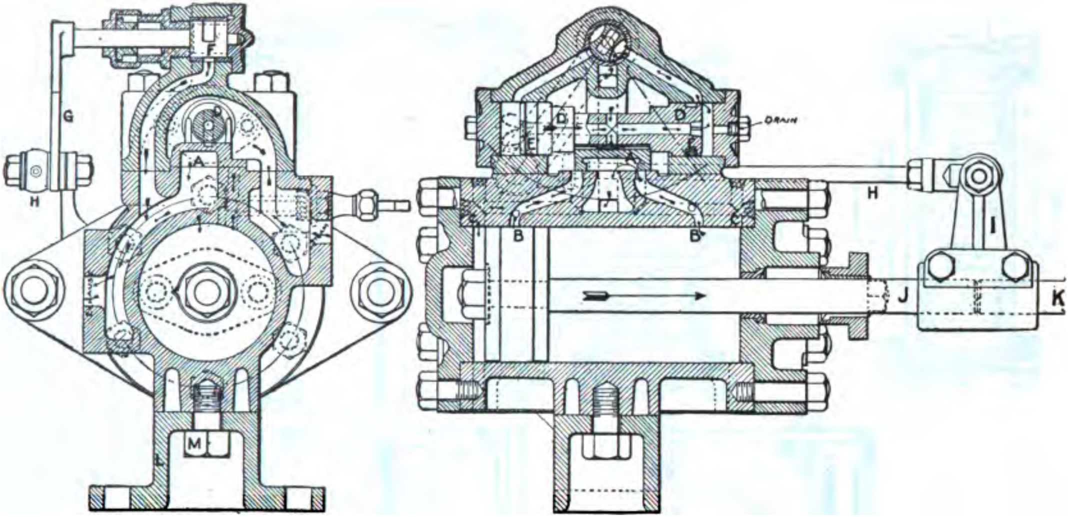
FIGS. 749 to 752.—*Continued.*

lever. When the piston rod moves, the lever rotates about P, carrying the tappet with it; and when the tappet strikes either collar it moves the movable seat in the direction in which the tappet is moving. By placing the collars so that the tappet strikes them before the piston reaches the end of its stroke, the movable seat will be shifted in the required manner.

To set the valves.—No valve adjustment is required to be made inside the steam chest, and the only adjustment which can be performed is that of altering the distance between the collars, thus changing the travel of the valve. This is done by loosening the set screws in the collars, and rotating the latter until they come to the required point. Changing the distance between the collars alters the length of the stroke. This is easily seen, because the action of the tappet in striking the collars is what admits and exhausts the steam; and if the distance which the tappet has to travel be varied, the time at which the valve is actuated is varied, and the stroke varies as well. The adjustment of these collars is very simple, and can be performed while the pump is running. In adjusting them it is desirable to make the stroke as long as possible and secure enough cushioning, for the shorter the stroke the greater the amount of the clearance, and the steam required to fill the clearance is wasted on every stroke. If the collars on the valve rod be not set at equal distances from the center line of the lever when the latter is vertical, the movable seat will be reversed sooner on one stroke than on the other, and consequently the piston will travel further in one direction than in the other.



Figs. 753 to 755.—Valve gear of the Davidson pump; an example of the *combined auxiliary valve and auxiliary piston* class. ***To set the valve piston,*** push the main pistons to the end of the stroke until the inner edge of the port and the piston coincide, then loosen the side lever, turn the cam C, until the valve piston uncovers the auxiliary steam port E, leading to the same end of the steam chest occupied by the main piston. ***After setting,*** secure the cam and then connect the side lever to the connecting rod. The side lever and cam occupy correct relative positions, therefore, the lever should be secured to the cam shaft while in this position. The stroke may be regulated by raising or lowering the end of the connecting rod in the slotted end of the slide lever. *Raising the connecting rod shortens the stroke and lowering it lengthens the stroke.* When making the foregoing adjustments it is well to have the connecting rod at or near the bottom of the slot as shown in the engravings.

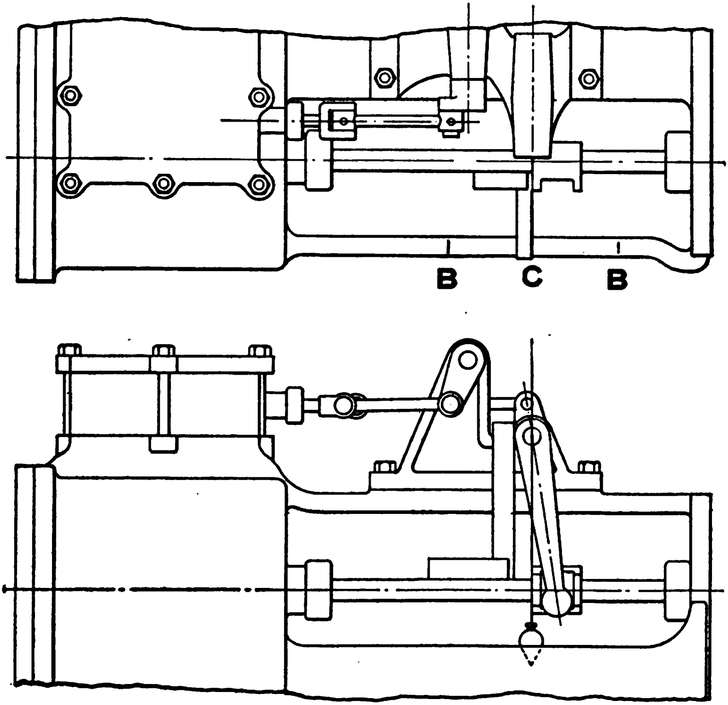


Figs. 756 and 757.—Valve gear of the *Smith-Vaile* pump. The **main valve A,** is a slide and is moved by a valve piston D. The supplemental valve F, has a reciprocating rotary motion which is communicated to it by the rock arm G, and pitman H, connecting with the cross head I, secured to the piston rod. It will be observed that the piston rod J, and the rod K, of the water end are separated so that should either one give out through wear or accident it can be replaced without sacrificing both, as would be the case if they were solid in one piece. This supplemental valve F, in general appearance very closely resembles the "Corliss" valve, and its action is somewhat similar, in controlling the action of the valve piston, which will be understood from the cut without further description.

cut off and compression, and fully opens the auxiliary ports E, to steam, and E_1, to exhaust.

By the admission of steam to one end, the other being open to the exhaust, the valve pistons move the valve to allow the admission and exhaust of steam from the cylinder for the return stroke. This main valve is as much under the control of the piston rod as is the valve of an ordinary steam engine worked by an eccentric which insures a positive action, the pump being capable of starting from all positions and maintaining a uniform and full stroke.

The Duplex Pump Valve Gear. If two steam pumps be placed side by side, it is found that the main valve of each may be operated directly from the piston rod of the other without the aid of any auxiliary pistons or valves as is necessary with single pumps. Each piston rod then with suitable connections operates the valve of the other pump in such sequence that the strokes are alternately made resulting in a discharge, nearer uniform than is obtained in a single pump. Figs. 758 and 759 show the general arrangement of the valve gear.

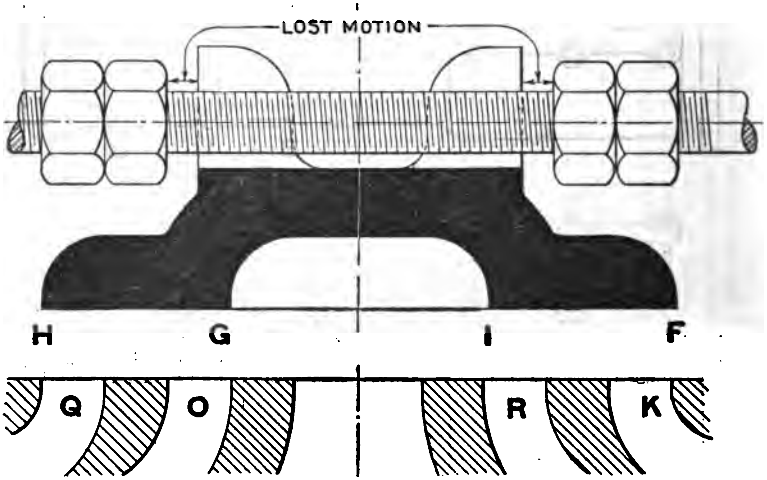


FIGS. 758 and 759.—Plan and elevation of one side of a duplex pump showing steam cylinder and valve gear.

The duplex pump has one main valve for each side, there being no auxiliary valves as in single pumps. These main valves, as shown in figs. 760 and 761, are nothing more than ordinary slide valves.

It will be seen from fig. 761 that the valve seat has five ports, giving separate steam and exhaust passages and a central exhaust cavity as shown.

The passages nearest the ends are steam passages while the inner passages are for exhaust. These inner passages are covered or closed by the piston just before the end of the stroke whereby a portion of the exhaust steam is compressed and made to act as a cushion between the piston and cylinder head, thus preventing the piston striking the cylinder heads when operating at high speed; this assists materially in the operation of the pump.



Figs. 760 and 761.—Main valve and valve seat of duplex pump; each "side" or pump is fitted with a valve and seat as here shown. H and F, are the steam edges of the valve and G and I, the exhaust edges. Q and K, are the steam ports and O and R, the exhaust ports; the exhaust cavity or outlet is seen at the center of the seat. Fig. 760, shows the lost motion between the stem and valve. The amount of lost motion given is such that the inlet ports are not closed and the exhaust ports opened too early in order to allow the piston to make a full stroke.

The travel of the valve is such that its exhaust edge never passes by the steam edge of the steam port, hence steam can only be exhausted through the exhaust port.

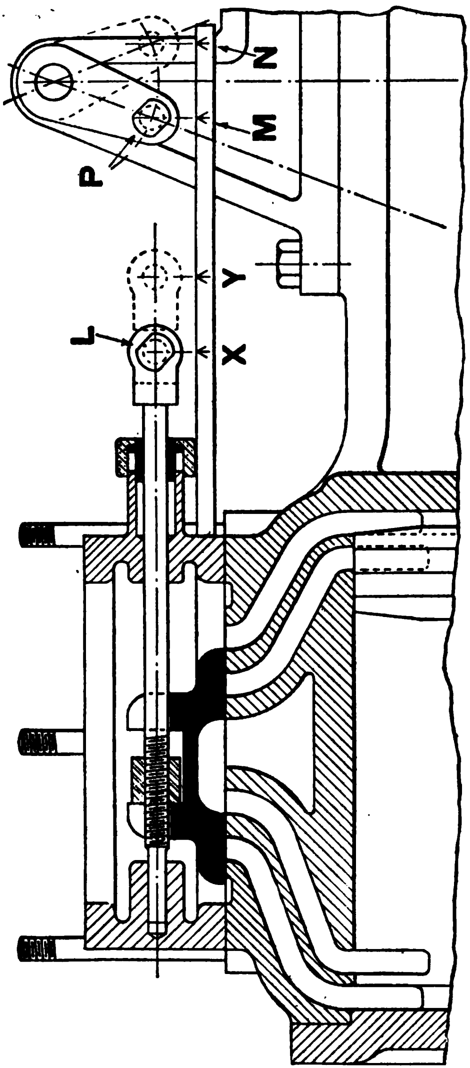


FIG. 762.—Valve and valve gear of duplex pump, showing rocker in its two extreme positions M and N, and the corresponding position of the valve stem end X and Y.

In large pumps, a by pass is provided, connecting the steam and exhaust ports; the by pass is provided with a stop valve by means of which the compression can be regulated. The adjustment of this valve depends upon the speed of the pump, the higher speeds requiring more compression.

The gear by which the motion of the piston rods is reduced and transmitted to the valves is fully shown in the accompanying cuts. From the illustrations it is seen that the piston rod of each cylinder is provided with a cross head which connects with a rocker arm. This rocker arm is attached to a rocker shaft, having at the other end a short rocker arm, which is connected with the valve stem of the other cylinder through a connecting link.

The valve stem is not rigidly attached to the valve but considerable lost motion is given, as shown in fig. 762, so that the valve is not moved until the piston has reached nearly half stroke.

Preliminary to

setting the valves of an old pump it should be ascertained if the cross heads have shifted on the piston rods.

Method of Locating Cross Head Centers.—Usually the cross heads of duplex pumps are held in position by a pin, which is driven through cross head and rod, after the former has been adjusted. It is impossible for the cross head to shift its position accidentally, unless the pin should drop out, and even then, there is a set screw, holding the cross head against slippage by ordinary use, and if such a thing should happen, the best way to readjust is to find its former position by the pinhole in the piston rod.

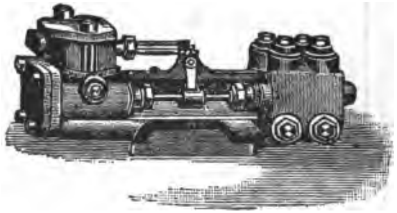


FIG. 763.—A very small Worthington duplex pump. Its dimensions are as follows: 2 inch diam. steam cylinder; 1⅛ inch water cylinder; 2⅔ inch stroke. Its capacity is .004 gallon per revolution; rev. per minute, 80; gallons per minute, 3.5. Steam pipe, ⅜ inch; exhaust pipe, ½ inch; suction pipe, 1 inch; discharge, ¾ inch. Floor space occupied, 1′ 9″ × 7″ wide.

Sometimes, however, it is necessary to replace the old piston rods by new ones, which may be quite frequently, if the water be bad, and steel rods are used. In most cases the engineer will find that the rods can be put into their proper places without any trouble, as the builders have always exact fitting duplicates in stock, but it is better to be sure of this, by making the following test: Mark the extreme position of the cross head on both sides of the pump on the frame or on a wood lathe, wedged in between the cylinder heads as shown in the figure.

If a lathe be not used, the positions of the cross heads may be transferred to the frame, by using a small set square.

Next put a mark, on either the frame or the lathe, to correspond with the central or mid-stroke position of the cross head. This may be obtained in two different ways, both being illustrated in figs. 758 and 759.

The use of the plumb bob as in fig. 759 should only be used if the pump

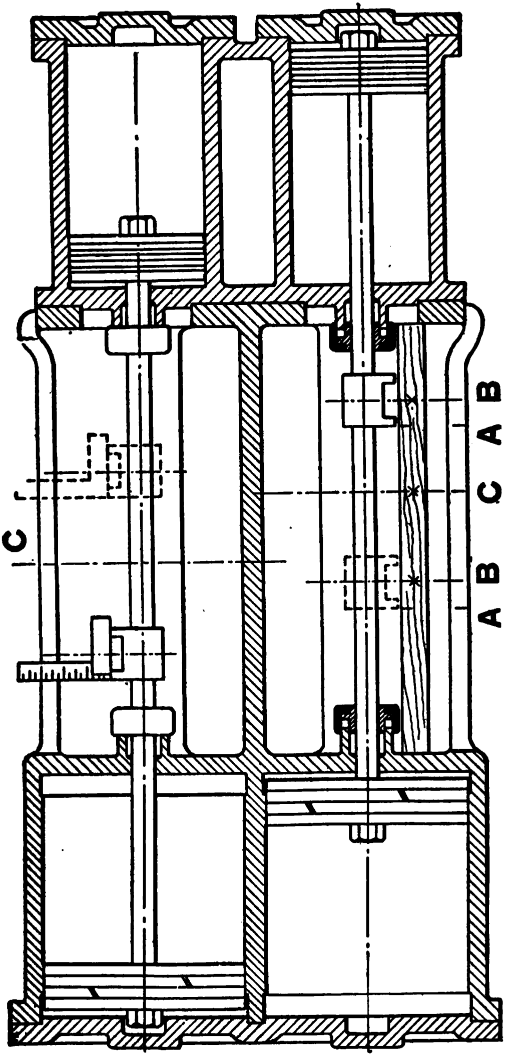


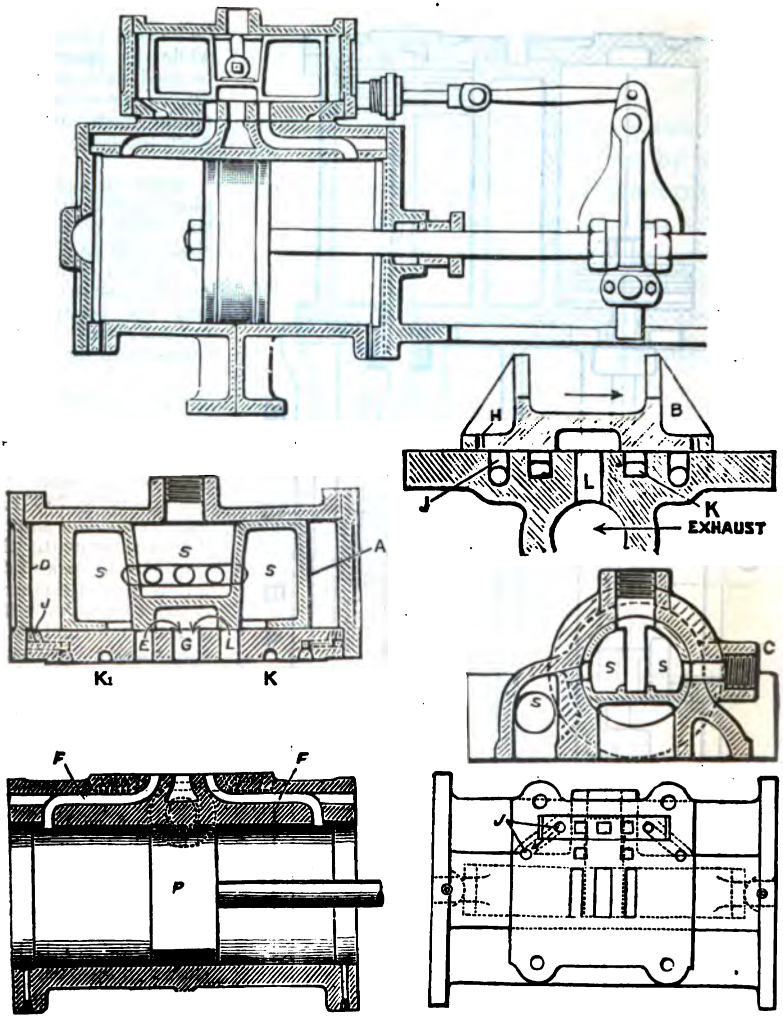
FIG. 764.—Location of cross head centers on duplex pumps.

be leveled properly, while the square may be used under any condition, providing the piston rods be not worn too badly.

When dropping the plumb bob in line with the center of the rock shaft as in fig. 759, the cross head may be moved close to the line, and its position be transferred to the frame as in fig. 758.

In the other method, the square is used against the hub of the rocker arm, and thus, it will be seen by examining fig. 759 that the heel of the square does not indicate the center of the rock shaft, but is out an amount, equal to one-half the diameter of the hub of the rocker arm, and the cross head should therefore not be set close to the square, but a distance, equal to the radius of the hub, away from it. This distance can be measured with an inside caliper, or a rule, and the position of the cross head is then transferred to the frame as in fig. 758, or marked on the lath as shown in fig. 764.

In both of the above methods, no marks have been made on



Figs. 765 to 770.—Valve gear of the ***Laidlaw-Dunn-Gordon*** pump. The admission of live steam to the cylinder and of exhaust steam to the atmosphere is controlled by a valve piston A, shown in fig. 766. Assume that the piston is in position shown, fig. 769, and that both the

the piston rod, which is always best to avoid, the cross head having served for a mark in both cases.

If the pump be small, there is no difficulty to move the pistons for this purpose, but on a large pump, the cross head may be unfastened, so as to be free to slide on the piston rod.

The marks A A, representing the extreme positions of the cross head have, however, been taken from one end of the cross head, and thus can not come equidistant from the mark C, representing the correct central position, even if the cross head be set correctly. Thus it will be necessary to transfer them toward the opposite end of the cross head, an amount equal to one-half the length of the cross head, B B, being the corrected marks.

If the position of the marks B B, be not equidistant from the center mark C, when the cross head is at the extreme ends of the stroke, it should be shifted on the piston rod, until in the proper position, the amount it is to be shifted will be indicated by the marks B B, fig. **764.**

It will not be necessary to shift the cross head on the rod, if it be out only a small amount, as the duplex pump is not such a sensitive machine, to require very delicate adjustment, and often it is found, that, if the

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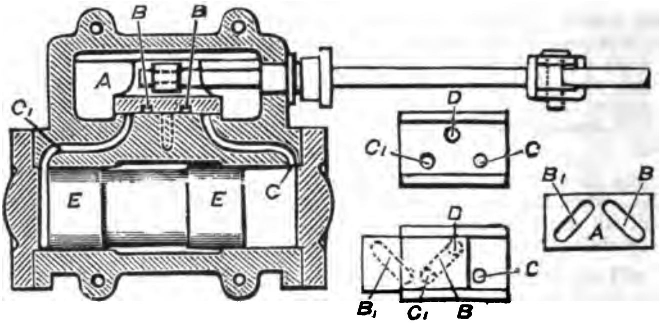
Figs. 765 to 770.—*Continued.*

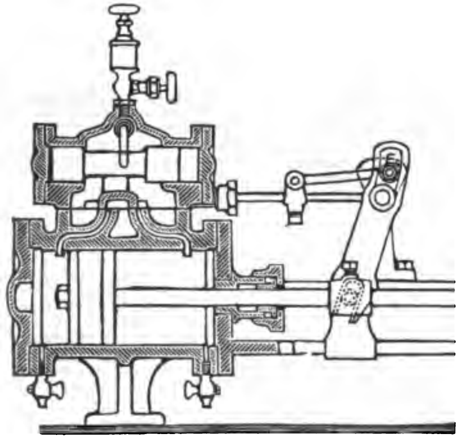
main and auxiliary valves cover their respective steam ports. By means of a starting bar, operating through a stuffing box in the valve chest, the piston valve A, is moved toward the head of the steam chest D, thus opening the ports E and L, and admitting live steam through L, from the cavities S, of the valve piston to the housing end of the main steam cylinder, through the port F fig. 769, forcing the main piston P, toward the opposite end of the stroke, or toward the left in the figure. The port E, fig. 453, being open, the exhaust steam escapes from front of the main piston through the port F, fig. 769, into the main exhaust port G, through the port E. The piston P, travels to its extreme left position and the auxiliary slide valve has been drawn to such a position in the direction indicated by the arrow in the smaller drawing in fig. 765, as to bring valve piston, A, toward the opposite end; the exhaust steam from the steam chest escapes from before it, through the exhaust port K, the opening of which into the chest is at such a distance from the head as will permit sufficient exhaust steam to remain to afford a cushion to the valve piston. With the auxiliary slide valve in position to bring the hole H, over the port J, fig. 770, it is plain that the exhaust through the port K, will pass into the main exhaust through the port L. With the main piston at its extreme travel toward the right, the ports E and L, which correspond to F and F, respectively in fig. 769, are opened in such a manner as to exhaust steam to the atmosphere from the housing end of the steam cylinder through the port F, and live steam from the chest to the head end of the main cylinder, through the port F, thus driving the main piston P, toward the housing end of the cylinder, or toward the right. The piston and reciprocating parts traveling in this direction move the auxiliary slide valve to its maximum point of travel in the opposite direction, thus opening the opposite auxiliary steam and exhaust ports and again driving the valve piston toward the head D, of the steam chest, whence a new stroke begins. Lost motion in the valve gear is taken up by adjustable links, on all sizes above 7 inches diameter by 10 inches stroke and on some smaller sizes. Cushioning of the steam pistons in the larger sizes and upwards is accomplished by means of suitable valves called cushion valves. In the smaller sizes sufficient cushioning is done by exhaust steam passing from the clearance space next the head through a small hole drilled into the main steam port.

To set the valve of this pump it is only necessary to place the piston in its central position and adjust the lever so that the valve will occupy its central position. By this proceeding the travel of the valve is equalized.

entire mechanism be set correctly, the pump will not work as well under steam, as if slightly out of adjustment.*

If the cross head be out of adjustment it is advisable to test the pump under steam, before making alterations. For this purpose the valves





FIGS. 771 to 775.—Valve gear of ***Dean Bros.*** pump. The auxiliary valve, A, fig. 771, has in its face two diagonal exhaust cavities, B, B_1. The ports, C C_1, and the exhaust port, D, are placed in a triangular position with one another, the diagonal cavities diverging so that the cavity B, when the valve is in place, connects the ports C_1 and D. Cavity B_1, connects the ports C and D, when the valve A, is at the end of the stroke. The three small cuts show relation of auxiliary valve to ports. The piston starts from left to right when the valve A, moves in an opposite direction, opens the port C, admitting steam to the auxiliary cylinder at the moment the main piston has reached the end of its stroke. The auxiliary piston E, is forced to the left, opening the main port and admitting steam to the main cylinder, reversing the movement of the main piston the return stroke of the main piston reverses the movement of the auxiliary valve, whereby the port C, is closed, at the moment the main piston reaches the end of its outer stroke. The port C_1, is opened by the valve A, and reverses the valve piston E, opens the main port and reverses the motion of the main piston. This port arrangement admits of a short valve with a long travel. The stroke of the pump can be regulated by moving the stud up or down in the segmental slot, should be adjusted to suit the original position of the cross head, and if possible, it will be found very useful, to attach a pointer to the crosshead, pointing toward that part of the frame, on which the center and extremes of the stroke have been marked.

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*NOTE.—If a pump work better when slightly out of adjustment it is due to irregularities in the steam and exhaust ports, and is liable to give more and earlier compression on one end of the cylinder than on the other, and, when running slow, the piston will not travel within the same distance from both cylinder heads.

By running the pump slow, it will be possible to ascertain the ends of the working stroke.

If the extreme positions of the pointer are marked, which can be done best by holding a lead pencil against the pointer, just touching the frame. The points, to which the lead pencil is pushed by the pointer on each end of the stroke, are the extremes of the stroke, when the pump is running, and by comparing these points with the marks previously obtained, indicating the true ends of the stroke, the clearance on each end can be obtained.

If there be considerable difference in the clearance on both ends, it is best to examine the valves, by moving the pistons by hand to the extremes of stroke, as found when running, and noting the port opening at both ends, for this purpose the valve chest cover has to be removed.

If there be any difference in port opening at both ends, this may be the cause of the unequal clearance, and a preliminary valve adjustment should be made, by equalizing the port opening approximately by eye.

Various types of pumps are provided with different means for such adjustments, but the principle remains the same, that is, to either lengthen or shorten the valve stems, as occasion demands.

Most all types of the smaller sizes are provided with the simple adjusting device, as indicated in fig. 762 which consists of a square nut, through which the valve stem is screwed, and by screwing the stem either in or out, it is respectively shortened or lengthened.

After such a preliminary adjustment has been made, the pump should

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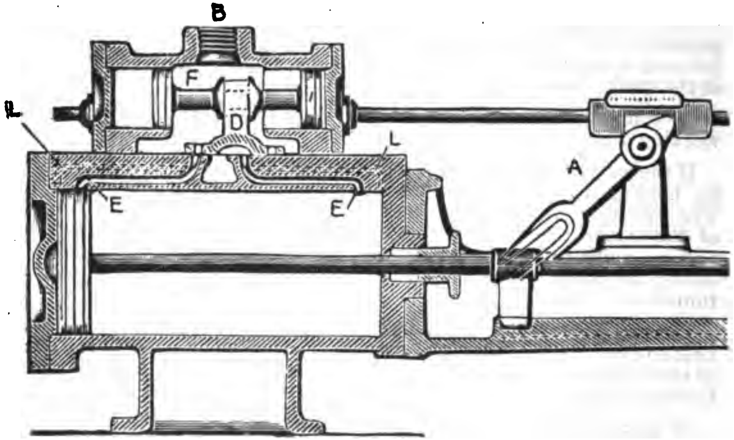
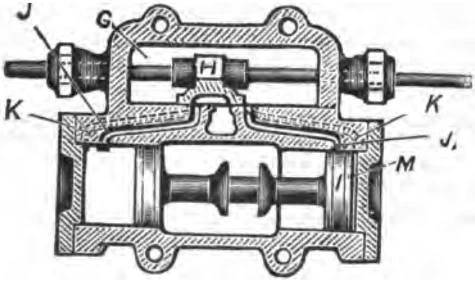
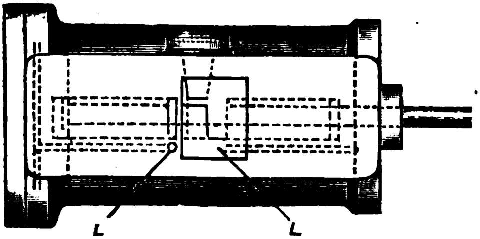
FIGS. 771 to 775.—*Continued.*

fig. 775, which varies the travel of the auxiliary valve and reverses the stroke of the main piston as desired. By raising the stud, the pump will make shorter strokes, and by lowering it, longer strokes.

To set the valve, turn the steam chest upside down. Put valve stem through the stuffing-box and secure in place the clamp for small slide valve. The diameter of valve stem is smaller where the clamp is attached. Now screw up the stuffing box nut (having previously removed the packing), then move the valve and stem so that the small port at right of valve will be open one-sixteenth inch and make a scratch upon the stem close to stuffing box nut. The valve should then be moved in the opposite direction to open the other small port one-sixteenth inch and make a second scratch upon the valve stem next to stuffing box nut. Prepare joint, and replace steam chest on cylinder. To square the valve, slacken the screw in cross head and move the latter to the end of stroke with edge of cross head flush with the end of guide, then set the valve stem so that the first scratch is flush with the face of nut, same as when the scratch was made. Tighten screw in set screw under valve rod dog and move the cross head to the opposite end of stroke, and note the position of second scratch. If it do not come to the position in which it was made, *split the difference by slackening the set screw under valve rod dog and move the valve rod to equalize the travel of valve.* In replacing steam chest on cylinder, cover the opening with a thin board, or piece of sheet iron, before turning it over to prevent the valve dropping out of place.







FIGS. 776 to 778.—Valve gear of the ***Burnham*** pump. Fig. 778 is a plan of the main cylinder valve face, having the same arrangement of ports as the cylinder shown in fig. 777. A longitudinal section of the steam cylinder is shown in fig. 776. Motion is imparted to the slotted arm and cam A, by means of a cross head and a roller on the piston rod. The cam works between and in contact with two blocks on the valve stem, and by adjusting these two blocks the stroke may be shortened or lengthened as the case may require. The valve stem of the auxiliary valve H, fig. 777, always moves in a direction opposite to that of the piston. The action of this valve alternately admits steam through the double ports J J and K K, to each end of the valve cylinder, causing the valve piston I, to move the main slide valve D, which, in turn, admits steam to the main cylinder through the double ports E E and L L. As the travel of the cam is only one-fifth that of the piston travel, the valve moves slowly, and without jar or noise which is often caused by long travel and rapid motion. Steam enters the steam chest at B, and fills the space F, between the valve piston heads and the auxiliary valve chest G, shown in fig. 776. With the auxiliary valve, H, in the position shown, fig. 777, steam passes into both ports J and K, but as the port J_1, is closed again be tried under steam, and the ends of the stroke should again be marked.

Should the marks denoting the clearance of the pistons again fall on the same points as before, and a difference in the clearance on both ends of the stroke be found, the trouble will be due to the irregular spacing of the ports in the cylinder bore, and there will be little chance for improvement, and, unless the cross head be found considerably out of adjustment, it should not be disturbed, and the final valve adjustment should be made to suit the extremes of the stroke while running.

It, however, rarely occurs that a pump is of such poor workmanship as to make proper adjustment impossible.

The location of the ends of the stroke does not make any difference in the manner of adjusting the valve, except, that it must be noted that in one case, by the end of the stroke, the extreme positions of the pistons when pried over, and in the other case the end positions of the pistons when allowed to run, are meant.

How to Set the Valves of a Duplex Pump.—Place a small stick or batten against the end of the valve chest, and mark the center of the pin P on the same, as indicated in fig. 762. Then move the piston, of the same side, to the other end of the stroke, and again mark the position of the pin P, on the same stick, as indicated by the dotted lines. The two marks M, and N, thus obtained, denote the extreme travel of the pin P.

It will now be necessary to obtain the marks X and Y on the same stick, which indicate the positions of the pin L when the valve has moved from one full port opening to the other.*

FIGS. 776 to 778.—*Continued.*

by the valve piston I, no steam can enter the valve cylinder through it, but the other port, K (extending to the extreme end of the valve cylinder), never being covered by the piston, is open, and admits steam into the space M. As this port is quite small the space fills slowly and the piston moves gradually until it uncovers the last port J_1, when the full volume of steam is admitted, which quickly moves the piston to the opposite end of the valve cylinder. During this movement of the valve piston, the large port J, remains open to the exhaust until it is covered by the valve piston. When the port J, is covered by the valve as at J_1, it has no connection with the exhaust, consequently, there being no outlet for the exhaust vapor, it is compressed and forms a cushion for the valve piston, I. The valve piston carries with it the main valve D, which admits steam to the main steam cylinder through the double ports E, E_1, and L, L_1, fig. 776. The same cushioning and slow starting of the piston occurs in the main as in the valve cylinder, each having double ports.

To set the valve.—Set the lever A, plumb and the valve to cover all the ports equally.

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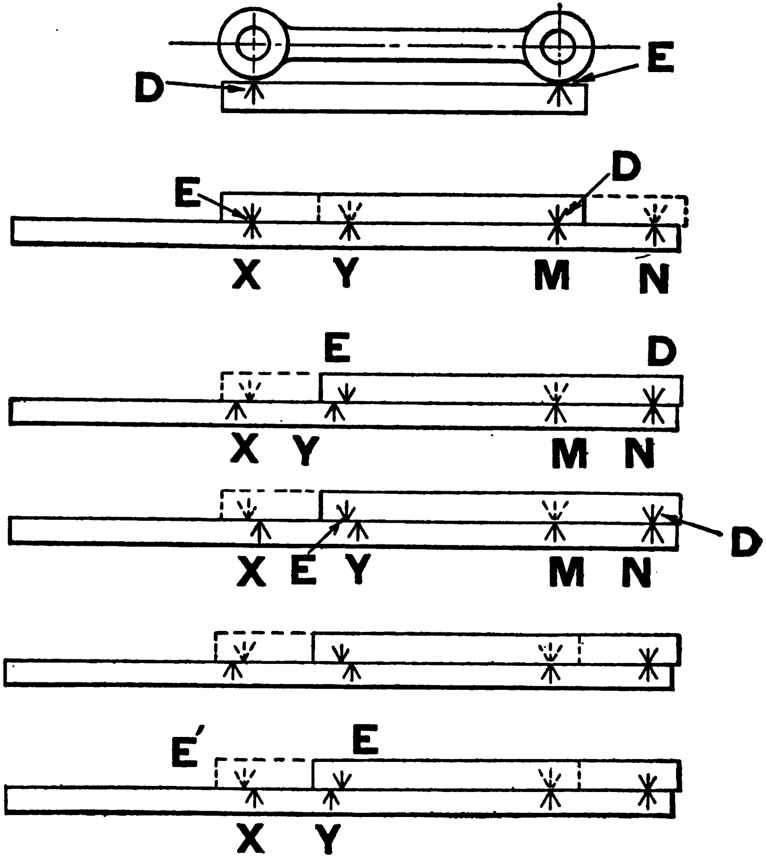
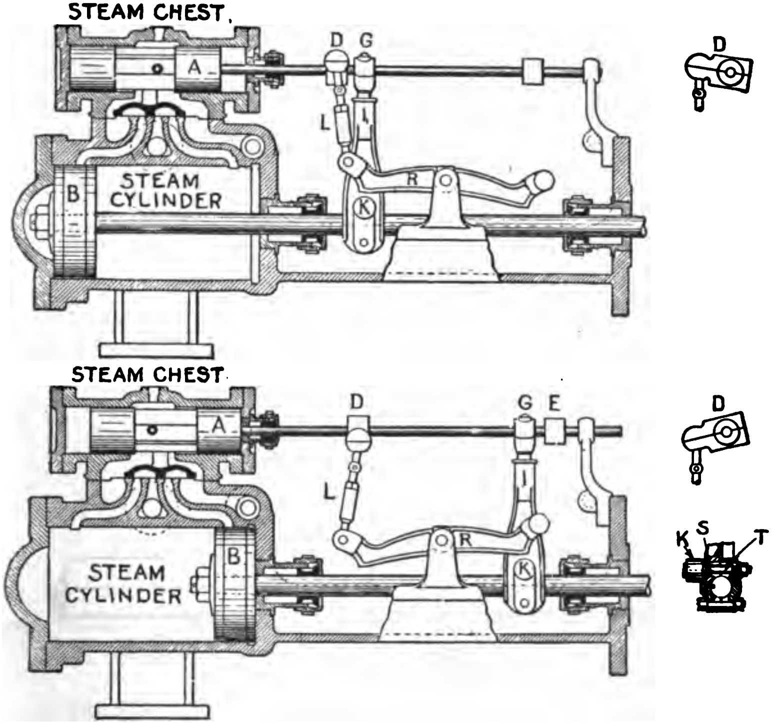
*NOTE.—When sliding the valve from one "full port" to the other, care should be taken to do this by moving the valve stem, to obtain the full effect of the lost motion between the nut and the lugs on the back of the valve as in fig. 760.

Now take a strip of stiff paper, and mark upon it the exact distance between the center of the holes in the valve connecting link, as D and E, fig. 779. Try the distance between the marks D and E, on the strip of paper, against the marks X and M, and Y and N, and if they should coincide as in fig. 780, the valve is correctly adjusted, and the links should be put into their places, and the valve chest cover replaced.



Figs. 779 to 784.—Paper template and batten with center marks as used in adjusting the valves of a duplex pump as fully explained in the accompanying text.

If, however, the marks should fall as in fig. 781, or fig. 782, it is evident that the valve stem is either too short as in fig. 781, or too long as in fig. 782,



FIGS. 785 to 789.—Valve gear of the ***Warren*** pump. ***To set the valve gear***: 1, Move piston valve A, and steam piston B, until they strike the heads of steam chest and steam cylinder, fig. 785. 2, Place clamp D, so as to allow ½ inch for Nos. 1, 2, 3, 4; ¾ inch for Nos. 5, 6, 6½; 1 inch for Nos. 7, 8, 9, 10, 11, 12, 13, five port; ½ inch for Nos. 7, 8, 9, 10, 11, 12, 13, three port, between clamp and tappet arm G, fig. 785, also between tip and collar E, fig. 787. 3, Set clamp D, so that as you roll or turn the valve rod in either direction as far as it will go, the clamp will be equally above and below the level. 4, Even up the motion of rocker R, by screwing the upper part of rocker connection L, out or in as required. 5, Set the roll K, in tappet arm I, up or down, so as to allow 3/16 inch between rocker and roll, when the latter is at its extreme of travel. Note the little set screw S, in roll stud T, fig. 789, which is adjustable to rest on bottom of tappet arm slot, and prevent the roll stud working down after it has been set in its proper position. 6, If, when the pump is run under steam, the tip G, strike clamp or collar violently before reversing its motion, the tappet arm roll K, needs to be raised, a little at a time, until such action ceases, otherwise the tip is liable to be broken. If, on the other hand, the pump run short stroke, drop the roll. The best adjustment is when the tip just misses hitting clamp and collar, when the pump is doing its regular work.

and it must be either lengthened an amount equal to the distance E Y, fig. 781, or shortened an amount equal to the distance E Y, in fig. 782.

Figs. 783 and 784, show other positions in which the marks on the stick and the strip of paper may fall. In both cases, the travel of the valve between the two inside edges of the steam ports evidently does not coincide with the travel of the pin P, fig. 762, indicating that there is either too much lost motion between the valve stem and the valve, as in fig. 783, or not sufficient, as in fig. 784. Before attempting to alter this, it is advisable to remove the valve entirely, and to see whether the distance between the steam and exhaust edges of the valve, as F and G, and H and I, fig. 760, correspond with the distances between the working edges of the ports K and O, and Q and R, respectively (fig. 761). If these distances agree with each other, and the marks representing the valve and pin travel fall as in fig. 783, it indicates that the valve has not sufficient motion to fully open the ports, hence less lost motion has to be given. Fig. 784 shows the reverse of this condition.

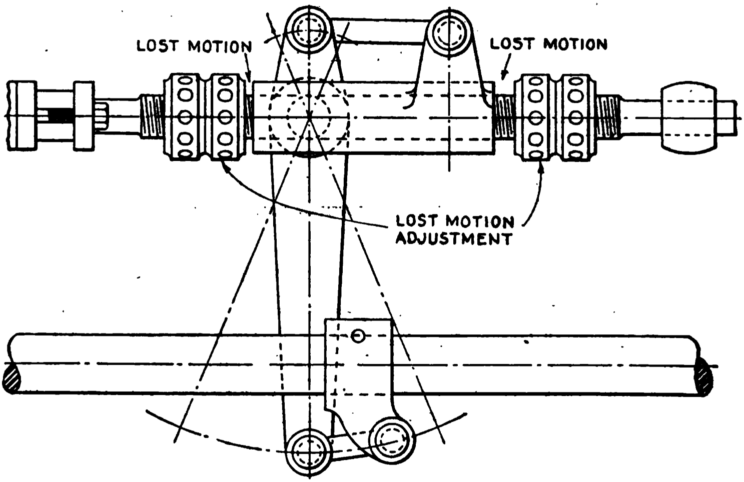


FIG. 790.—Rocker arm and connections as usually designed for large pumps, with lost motion adjustment. In this arrangement, the lost motion can be adjusted while the pump is in operation.

Should the distance, between the edges F and G, or H and I, be found shorter than the distance between their respective port edges, an amount equal to one-half the difference between E′ E and X Y, fig. 784, the steam edge of the valve will over travel the inner edge of the steam port, when the valve is connected up, but the exhaust port would have just full opening, indicating that there is some exhaust lap, and if the pump be found to run smooth, it is advisable not to tamper with the adjustment of the lost motion.

If it be necessary to increase the lost motion between valve and stem, on a pump provided with such an adjustment as in fig. 762, it can be done by decreasing the width of the nut, by filing or machining in a shaper.

To decrease the lost motion, either a new nut must be provided, or sheet metal washers of the required thickness may be cut, and placed on the valve stem between the nut and the lugs on the back of the valve.

The method of valve setting just described is only suitable for small pumps; the larger ones generally being provided with an adjustment as in figs. 760, 761 and 791. The arrangement shown in figs. 760 and 761, is very simple, and permits accurate adjustment, but in order to do this, it is necessary to remove the valve chest cover.

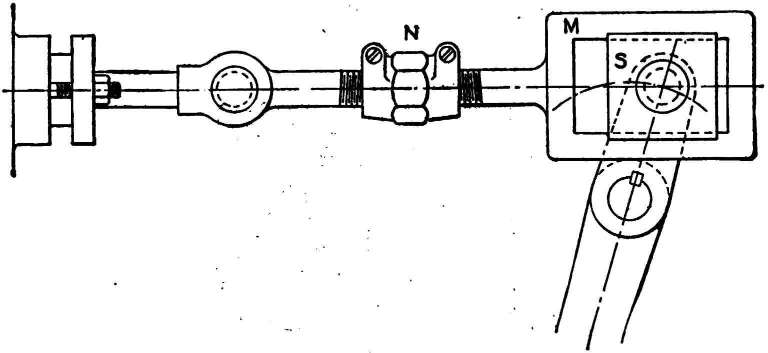


Fig. 791.—Lost motion arrangement consisting of yoke M, and block S, pivoted at the rocker end. In this design the lost motion cannot be changed without altering the length of the block S, but the length of the valve stem can be adjusted by means of the sleeve nut N.

The type shown in fig. 791, is mostly used on large and more expensive pumps, and permits alterations in the adjustment being made while the pump is running.

In fig. 791, the lost motion can not be altered, without taking off or adding to the ends of the block S, but the sleeve nut in the connecting link is a good device for altering the length of the valve stem.

It is seldom the case that the amount of lost motion has to be altered, and unless the operator be thoroughly familiar with the details and design of the pump, he should not undertake such alterations, as the designer knows best what the requirements are.

The above directions can not always be closely followed, as the different designs require different treatment, but by thoroughly understanding the above, the beginner will be greatly assisted even with the most complicated construction.

Short Rules for Setting the Valves of a Duplex Pump.— It may be helpful in acquiring a knowledge of how to set the valves

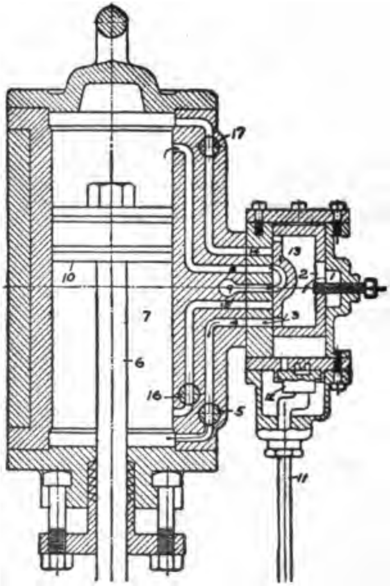


FIG. 792.—Sectional view of *American* deep well pumping head showing valve. In the position shown the steam is passing through ports 1, 2, 3, 4, and choke valve 5, into the cylinder 7. Piston rod 6, is connected to the pumping rods that extend down into the well to the water plunger. The number of strokes is regulated by the choke valves 5, 16 and 17. Port 8, is opened for the exhaust of steam out through 9. Just before the piston 10, reaches the end of its stroke, it closes the exhaust port 8, and forms a cushion. At the same time the valve stem 11, is also turned by a roller on the cross head, striking a finger cam on the valve stem. The movement changes the position of the auxiliary valve 12. Steam will then flow through suitable ports and will move the valve 13, so that port 14, is uncovered, allowing steam to enter the upper end of the cylinder and cause the piston to move in the reverse direction. At the same time port 15, is brought into communication with exhaust 9. Choke valve 16, controls the exhaust and helps the regulation of the pump. In operating a single acting cylinder, where the weight of pump rods is heavier than water, no steam is used on the down stroke; then valve 17, is closed and valve 16, shut sufficiently to sustain the weight of the pump rods and give the required number of strokes. When the double acting and two stroke cylinders are used, valve 17 is opened sufficiently to give a uniformity to up and down strokes.

to consider simply the essential operations without the various details or methods of performing them as given in the foregoing instructions: They may be briefly expressed in the form of rules as follows:

1. *Locate the steam piston in the center of the cylinder;*

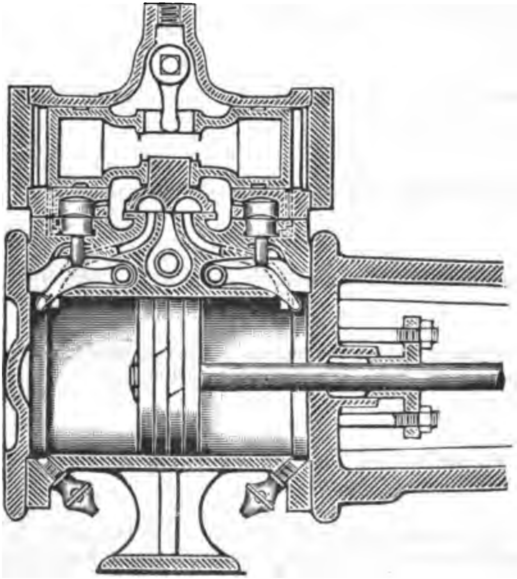


FIG. 793.—Valve gear of the McGowan pump. Its main valve is of the B form and is driven by a valve piston. Steam enters the central port in valve seat and into the cylinder through one of the cavities in the valve and exhausts through the opposite. The two tappet valves cover the auxiliary ports, shown by dotted lines, leading to the ends of the steam chest and connect with the main exhaust ports. When the piston reaches the end of its stroke it lifts one of the tappet levers and with it the corresponding valve is raised from its seat, opening the port leading from the end of the steam chest to the main exhaust port. The pressure is thus relieved on one end of the valve piston and the steam pressing on the opposite end forces the valve piston to the opposite end of its stroke, thus reversing the distribution of steam to the cylinder and starting the piston on its return stroke. The main valve is connected with the valve piston so that all lost motion is taken up automatically. A rocker shaft, extending through the steam chest carries a toe moving in a slot in the top of the valve piston, so that the valve can be moved by hand.

To set the valves.—Simply keep the valves in order. The motion of the piston as it nears the end of the stroke opens and closes the valves.

This is accomplished by pushing the piston to one end of its stroke against the cylinder head and marking the rod with a scriber at the face of the stuffing box, and then bringing the piston in contact with the opposite head.

2. *Divide exactly the length of this contact stroke;*

Shove the piston back to this half mark; which brings the piston directly in the center of the steam cylinder;

3. *Perform the same operation with the other side;*

4. *Place the slide valves in their central position;*

5. *Pass each valve stem through the stuffing box and gland;*

The operation of placing the pistons in the center of their cylinders brings the levers and rock shafts in a vertical position;

6. *Screw the valve stem through the nuts;*

The stem is screwed until the hole in the eye of the valve stem head comes in a line with the hole in the links, connecting the rocker shaft.

7. *Put the pins in their places;*

8. *Adjust the nuts on both sides of the lugs.*

Leave about one-eighth to one-fourth inch lost motion on each side.

CHAPTER 13

VALVE SETTING

How to Set the Slide Valve.—In the ordinary valve gear, such as the type shown in fig. 794, the eccentric is retained in position on the shaft by a set screw, and the length of the valve stem made adjustable by a threaded end with jamb nut, or equivalent. The valve stem may, therefore, be lengthened or shortened, and the eccentric placed in any angular position.

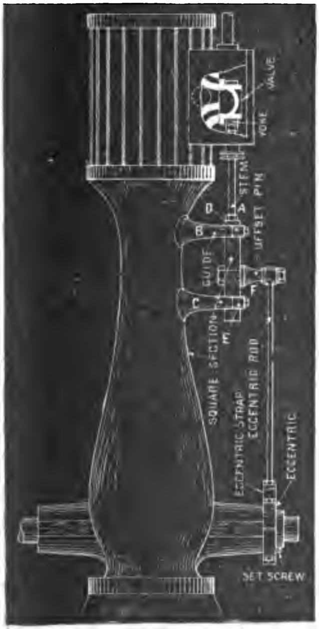


Fig. 794.—Plain slide valve engine with ordinary valve gear; a type of engine well adapted for practice in valve setting.

On assembling the valve gear it is found that *the dimensions for the valve stem length, and eccentric position are* **lacking.**

In setting the valve there are three distinct operations which are to be performed in the order here given:

1. *Locating the engine on the dead centers;*

2. *Finding the length of the valve stem, that is, equalizing the lead;*

3. *Determining the correct position of the eccentric.*

How to Find the Dead Center.—The engine is located on the dead center with a tram, such as shown in fig. 795.

This consists of a piece of one-fourth inch or three-eighth inch tool steel rod, of suitable length corresponding to the size of the engine and having a small portion at one end bent to a right angle; each end being ground to a fine point and hardened. The tram, in fact, corresponds to the bent scriber of a scribing block.

A permanent center punch mark is made on the engine frame to receive the straight end of the tram, and a ring of small punch marks made around this permanent mark, to easily identify its position for future occasions, especially after painting.

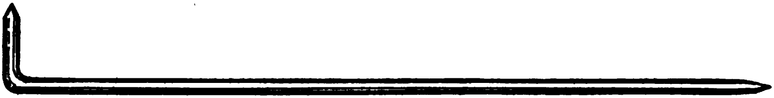


FIG. 795.—Tram, or instrument used in finding the dead center of an engine. It corresponds to the bent scriber of a machinist's scribing block.

On a vertical engine, the permanent mark may be located on the column or bed plate; on a horizontal engine, on the bed plate, in either case the punch mark should be made at some convenient place where the other end will reach to the crank disc, or fly wheel.

The dead center may now be located as follows: The engine is turned *in the direction in which it is to run* until the piston has nearly completed the stroke, as shown in fig. 796, crank position B. A mark M, to indicate this position, is made across the guide and cross head. With the straight end of the tram in the permanent punch mark P, as a center, an arc C, is described on the side of the fly wheel rim, the surface first being cleaned of oil, and rubbed with chalk so the mark is easily seen. The engine is now turned *past* the dead center until the mark on the guide again registers with the mark on the cross head corresponding to crank position A. Arc D, is now described with the same center P, and an arc passing through C. and D, is described from the center of the shaft. That portion of the arc included between C and D, is bisected, giving the point E. A punch mark is made at this point and the engine turned *in the direction of its future rotation* until E, registers with the bent end of the tram when its other end is in P. In this position the engine is on the dead center. The other dead center is found in a similar manner.

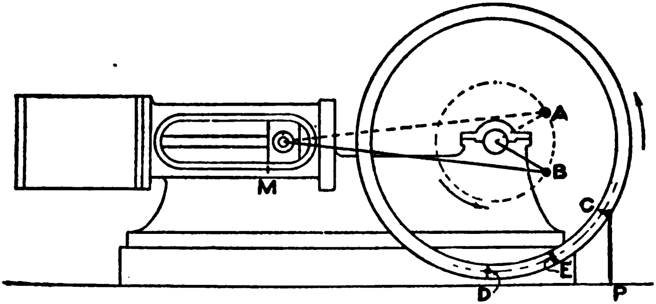
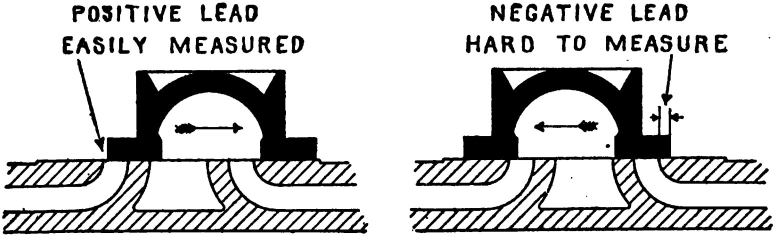


FIG. 796.—Locating the engine on the dead center. In doing this as described in the text, the engine should always be turned in the direction in which it is to run so as not to introduce any error due to lost motion. The tram marks should be made permanent with a center punch.

The engine should always be turned in one direction in order not to introduce any error due to lost motion in the wrist and crank pins. It matters not which direction is followed, the object being to have the crank pin pressing against the same brass for each adjustment. It is usual, however, to turn the engine *in the direction in which it is to run*, presumably because this is more easily remembered.

In case the engine has been moved too far at any time, it is not necessary to complete the revolution, but merely to turn it back beyond the desired point, and then forward again up to that point in the direction of rotation thus taking up the lost motion each time in the same direction.

Adjusting the Valve Stem.—Having located the dead centers, the second step is to *equalize the lead*, that is, to make it the same at each end of the cylinder. First, the eccentric is located "by eye," placing it *ahead* of its correct position rather than behind.



FIGS. 797 and 798.—Positive and negative lead. In setting a valve the eccentric is first located "by eye." The figures show why it should be placed ahead of its correct position rather than behind.

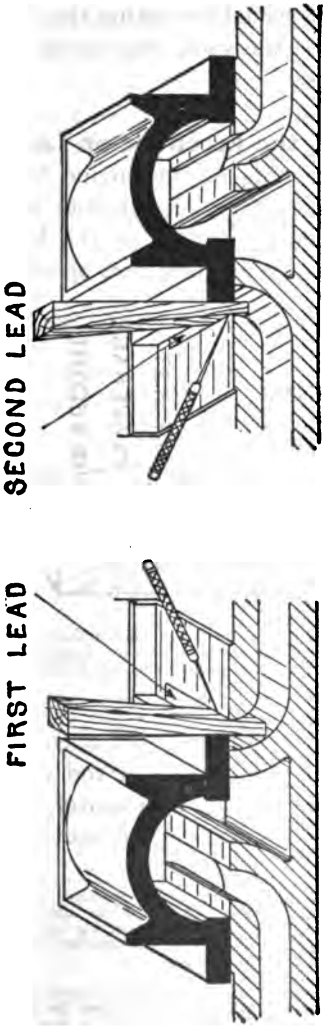
With the eccentric set ahead, the lead is *positive* at each end; when behind, it will probably be *negative* at one or both ends, that is, the two lead positions of the valve would be about as shown in figs. 797 and 798. The reason for setting the eccentric ahead is to avoid negative lead as in fig. 798, because it is not easily measured.

A long wooden wedge should now be provided, tapering from one-half inch (more or less depending on the size of the engine) down to "nothing," and cut into several pieces as shown in figs. 799 to 801.

FIGS. 799 to 801.—Wedges for measuring lead. Prepared from a long piece of wood, tapered from one-half inch (more or less depending on the size of the engine) down to "nothing" and cut in several pieces.

With eccentric set well advanced, the engine is placed on the dead center, and the amount of lead measured by one of these wooden wedges, after which the lead for the other end is found in a similar manner.

In measuring the leads, a suitable wedge is inserted into the ports as far as it will go, as shown in figs. 802 and 803, being careful to keep one side of the wedge in contact, that is, parallel with the end of the valve, and perpendicular to the seat. For each end, a line is scribed across the wedge along the steam edge of the port, as shown in the figures; these lines (A and B, fig. 804) indicate the lead at the two ends, being located at points on the wedge where the thickness is equal to first and second leads. A line C, drawn half-way between A and B will represent the *average*, or equalized lead. In taking lead measurements with a wedge the same side should, of course, always be placed next to the valve.



FIGS. 802 and 803.—Equalizing the lead. By means of a wedge and scriber, the lead is measured at each end of the cylinder, and the average found as in fig. 804. In using the wedge the same side must always be placed next the valve, and held true to avoid error. Why is this?

The lead may now be equalized by adjusting the length of the valve stem so that the wedge will enter the port up to the line of average lead (C, fig. 804). If the work has been correctly done, the lead will be the same at each end.

Finding the Correct Position of the Eccentric.—Since the eccentric was set "by eye," the lead is probably too great, or too small as the case may be. To correct this, the eccentric is turned on the shaft, *in the direction in which the engine is to run*, until the valve has the desired lead.* The results should be verified by testing the lead at the other end, and if both leads be the same, the valve has been correctly set.†

Finding the Correct Position of the Eccentric on Large Engines.—To avoid the frequent turning of the engine from one dead center to the other, the necessary adjustments may be made by equalizing the port opening instead of the lead. The eccentric is turned until it gives the maximum port opening, first at one end, and then at the other. These port openings, if unequal, are equalized by adjusting the length of the valve stem, after which, the engine is placed on the dead center, and the eccentric turned until the valve gives the desired lead.

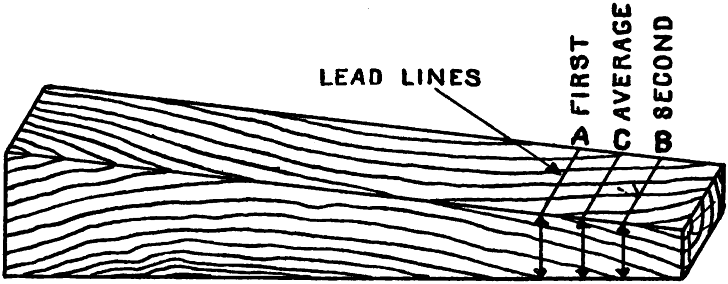


FIG. 804.—Equalizing the lead. After obtaining the lead lines A and B, as in figs. 802 and 803, a line C, is scribed half way between, which gives the average lead. The valve stem is then adjusted so that the wedge will enter the port up to C, thus making the lead the same at each end of the cylinder.

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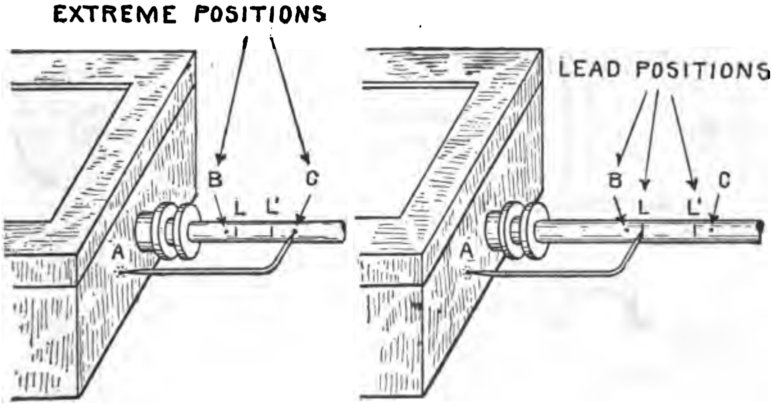
*NOTE.—The lead given to engines varies considerably from a small negative lead to three-eighths inch or more positive lead, depending on the type and size of the engine; its amount is decided upon arbitrarily by the designer but may be varied in setting the valve simply by changing the angular advance of the eccentric. In general, the amount of lead depends on the speed of rotation, and the inertia of the reciprocating parts.

†NOTE.—In order to clearly fix in mind the general principles involved in setting a slide valve, it is recommended that the instructions be read a second time, omitting the minor details given in the small type, as these tend to divert the attention from the important operations.

Setting the Slide Valve Without Removing the Steam Chest Cover.—If the set screw of the eccentric should work loose during operation, and the eccentric change its position, it may be quickly reset without taking off the steam chest cover, thus saving valuable time in case of a shut down.

A permanent punch mark is made on the end of the steam chest as at A, fig. 805, for taking measurements on the valve stem with a tram. The eccentric is fastened in any position, and the engine turned until the valve is at one end of its travel. A tram mark B, is made on the stem to indicate this position, and similarly another mark C, to indicate the other end of the valve travel. The engine is placed on each dead center and the position of the valve located by the tram. This gives the two linear advance positions L and L′, of the valve, which in case the eccentric has been incorrectly set, are at unequal distances L B and L′C, from B and C.

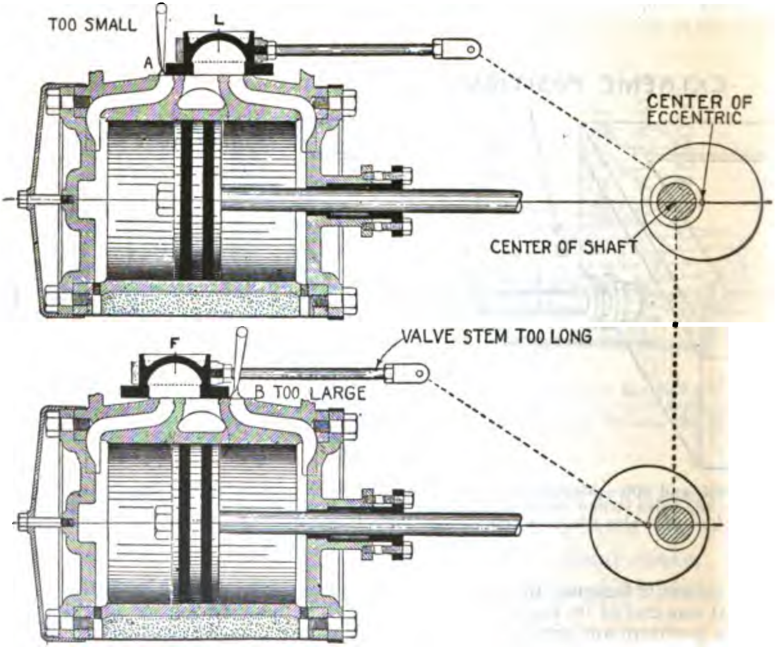


Figs. 805 and 806.—Method of setting the valve without removing the steam chest cover, when the valve stem does not require adjustment. It consists in equalizing the two positions L, L′, of the valve when the engine is on the corresponding dead center.

It remains to adjust the position of the eccentric until L B, is equal to L′ C, as shown in fig. 806. With this condition fulfilled, the valve is correctly set.

Setting the Slide Valve Without Finding the Dead Centers.—In the case of a very large engine where the operation of putting the engine on the dead centers would require one or more assistants, the following method will be found useful. It consists of:

1. Equalizing the port opening;
2. Finding the angular advance.



Figs. 807 and 808.—Setting the valve without putting engine on the dead center: 1, ***equalizing the port opening.***

Equalizing the Port Opening.—For this purpose a pair of inside calipers may be used when the port opening exceeds the width of the port, or if less, a wedge should be used.

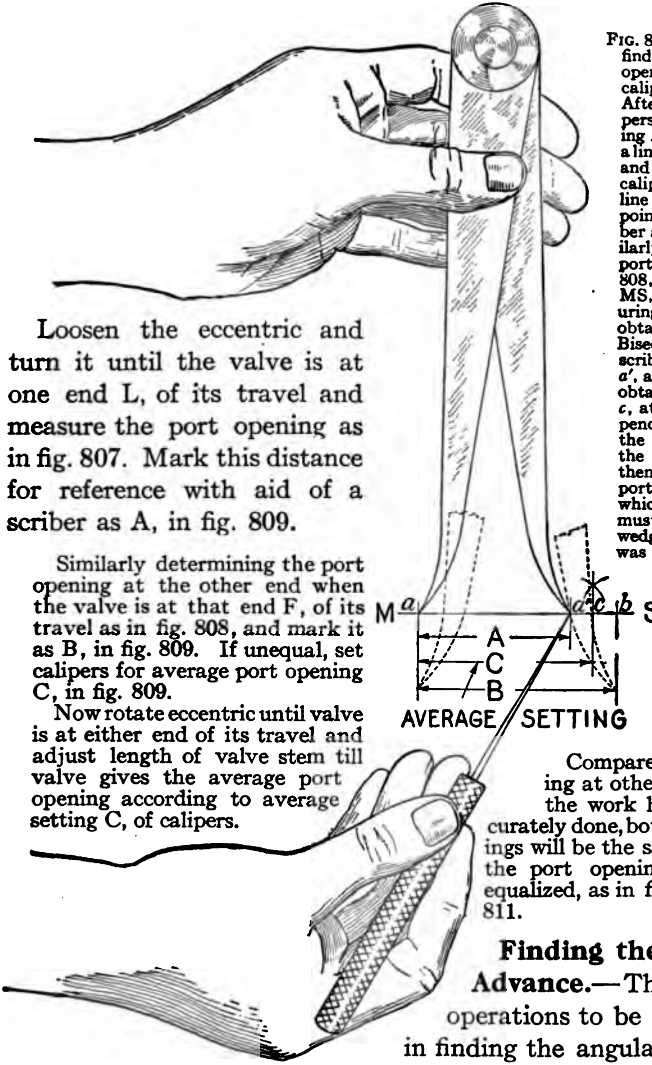


FIG. 809.—Method of finding average port opening with inside calipers and scriber. After setting calipers for port opening A, fig. 807, draw a line as MS, fig. 809 and place ends of caliper legs on this line; mark these points *aa'*, with scriber as shown. Similarly, set calipers for port opening B, fig. 808, and transfer to MS, fig. 809, measuring from *a*, and obtaining point *b*. Bisect *a'b*, by describing arcs about *a'*, and *b*, as centers, obtaining the point *c*, at foot of a perpendicular through the intersection of the arcs, *ac*, or C, then is the average port opening for which the calipers must be set. The wedge method as was used for equalizing the lead can be used to advantage when the port opening is less than the width of the port.

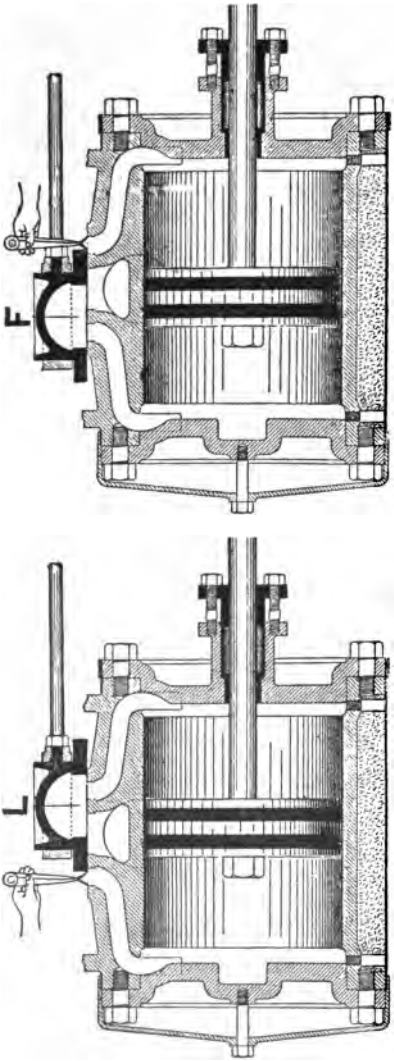
Loosen the eccentric and turn it until the valve is at one end L, of its travel and measure the port opening as in fig. 807. Mark this distance for reference with aid of a scriber as A, in fig. 809.

Similarly determining the port opening at the other end when the valve is at that end F, of its travel as in fig. 808, and mark it as B, in fig. 809. If unequal, set calipers for average port opening C, in fig. 809.

Now rotate eccentric until valve is at either end of its travel and adjust length of valve stem till valve gives the average port opening according to average setting C, of calipers.

Compare port opening at other end and if the work has been accurately done, both port openings will be the same, that is, the port opening has been equalized, as in figs. 810 and 811.

Finding the Angular Advance.—The several operations to be performed in finding the angular advance and placing tne eccentric in angular advance position are:



Figs. 810 and 811.—Setting the valve without putting the engine on the dead centers: *1. the port opening equalized.*

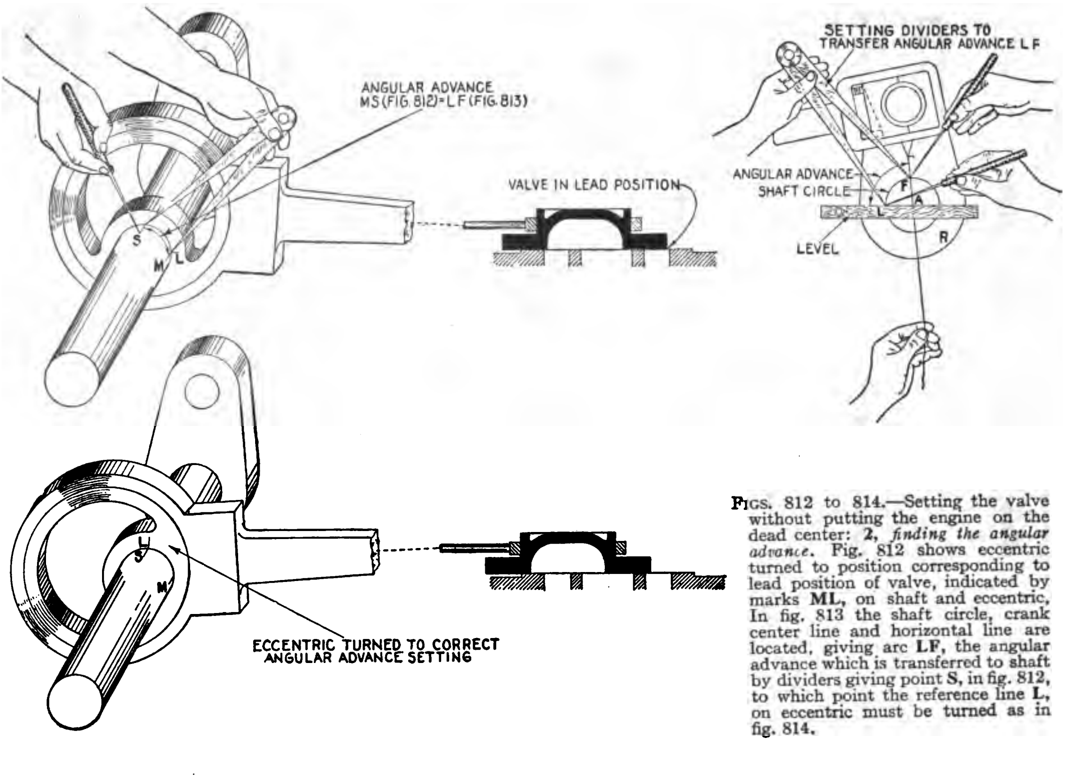
1. Rotating eccentric till valve is in lead position;
2. Finding angular advance;
3. Transferring angular advance to reference mark on shaft;
4. Rotating eccentric to angular advance position.

Performing these operations in the order given, first rotate eccentric until the valve has the desired lead and locate this position of the eccentric by scribing a line **M,** on the eccentric and **L,** on the shaft as in fig. 812.

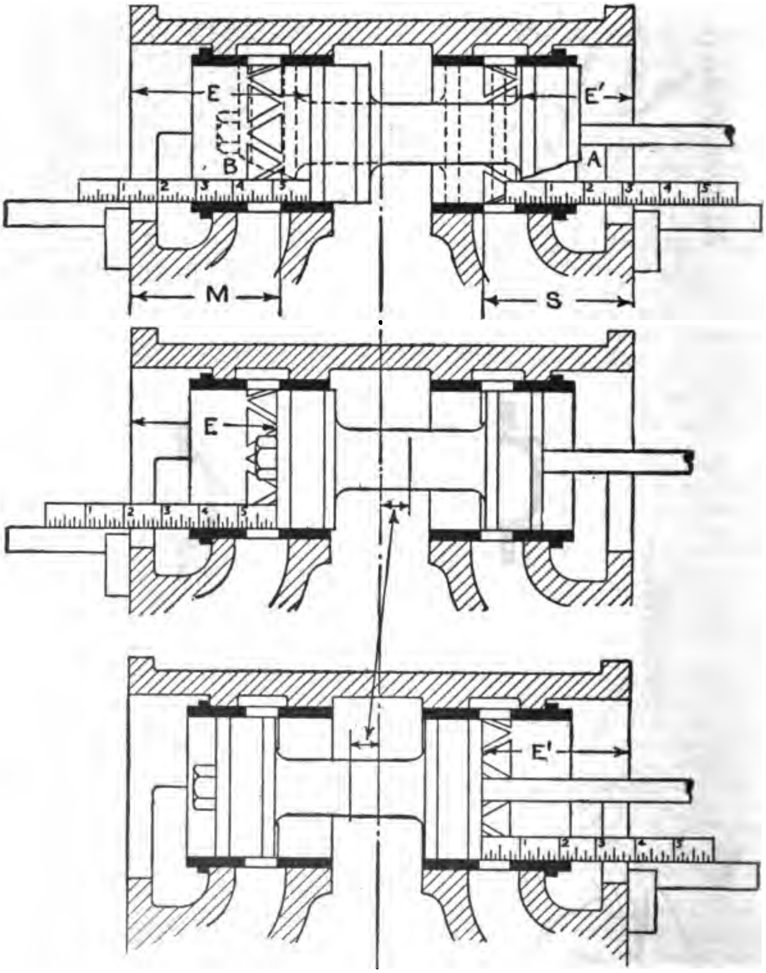
Now measure diameter of shaft at eccentric with outside calipers and set dividers to half this length. Wipe all grease from the crank and chalk same. With one end of the dividers in the lathe center scribe a circle R, corresponding to the shaft diameter as in fig. 813.

Next take a small string, or preferably strong flax thread and make a loop at one end and place it around the crank pin; pull very taut, holding it so that it intersects the shaft lathe center, and scribe the point **F′,** where it cuts the shaft circle. The thread then represents the center line of the crank.

Now by means of a level, scribe a horizontal line passing through the shaft center and cutting the shaft circle at **L,** *Evidently the angle* **LAF,** *is the angular advance,* which can be easily transferred to the eccentric by a pair of dividers.



FIGS. 812 to 814.—Setting the valve without putting the engine on the dead center: 2, *finding the angular advance*. Fig. 812 shows eccentric turned to position corresponding to lead position of valve, indicated by marks **ML**, on shaft and eccentric, In fig. 813 the shaft circle, crank center line and horizontal line are located, giving arc **LF**, the angular advance which is transferred to shaft by dividers giving point **S**, in fig. 812, to which point the reference line **L**, on eccentric must be turned as in fig. 814.



FIGS. 815 to 817.—Setting an inside admission piston valve. The two extreme positions are measured as in fig. 815, and the travel equalized by adjusting the valve stem. The engine is placed on each dead center, and measurements taken as in figs. 816 and 817. The position of the eccentric is then adjusted until E, fig. 816, equals E', fig. 817. If M and S be unequal due allowance should be made.

Set the dividers as in fig. 813 to measure the arc M′ S, then place one leg of the dividers on M, in fig. 812 and lay off M S, in the direction of rotating.

Turn the eccentric till its reference mark M, coincides with S, as in fig. 814 and secure eccentric in this position, when, if all the operations have been properly performed the setting will be found correct.

Setting an Inside Admission Piston Valve.—In this type of valve the steam edge of the port being on the inside, the valve cannot be set by direct lead measurements as with the slide valve; use is therefore made of the exhaust edge of the valve as a basis for measurements. The necessary operations in setting the valve should present no difficulty if the two following principles be understood and remembered:

1. *The two extreme positions of the valve must be equally distant on either side of the neutral position.*

2. *With equal lead, the linear advance must be the same for each end of the cylinder,*

Applying the first principle, the valve gear is adjusted so that the valve travels an equal distance each side of its neutral position.

To do this, the eccentric is set in any position on the shaft, and the engine turned over until the valve has reached one end of its travel as position A, fig. 815. The distance E, from the exhaust edge of the valve to the end of the cylinder, is measured, and similarly, distance E′ when the valve is at the other end of its travel, as in position B, shown in dotted lines. If the length of the valve stem be not correct, these two distances will be unequal. The travel is now equalized by adjusting the length of the valve stem until these distances become equal, that is, until E = E′.

Applying the second principle, the engine is placed on each dead center, and the distances of the exhaust edges of the valve from the ends of the cylinder measured.

If the eccentric has not the proper angular advance, these distances will be unequal, and it remains to adjust the position of the eccentric until they become the same, as shown in figs. 416 and 417.

When E, fig. 816 is equal to E', fig. 817 the linear advance is the same for each end of the cylinder, hence the lead has been equalized and the valve correctly set.

Before setting an inside admission valve as just outlined, the location of the ports with respect to the cylinder ends should be carefully determined. In most cases these are equidistant from the ends, that is, M = S, fig. 818; if not, due allowance should be made in setting the valve.

In locating the ports, the measurements are conveniently made with an ordinary rule having a strip of metal soldered on the brass end as shown in fig. 818; in setting the valve, a short steel rule is used. In either case a square or straight edge is used in taking the readings to project the plane of the cylinder to the rule as shown in the figures. The rule must be held plumb with the valve to avoid error.

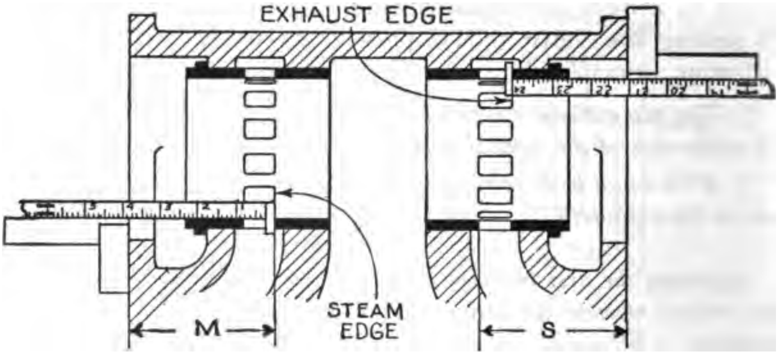


FIG. 818.—Method of measuring the location of the ports. A strip of metal is soldered on the brass end of the scale as shown, being placed at the side to measure the steam edge M, and over the end, for the exhaust edge S.

Emergency Rule for Setting the Slide Valve.—If the eccentric should slip on the shaft, or any other accident throw the valve gear out of position, it may be quickly reset as follows: The engine is placed on the dead center and the eccentric turned a little *behind* its correct position.

With the cylinder drain cocks open, a small amount of steam

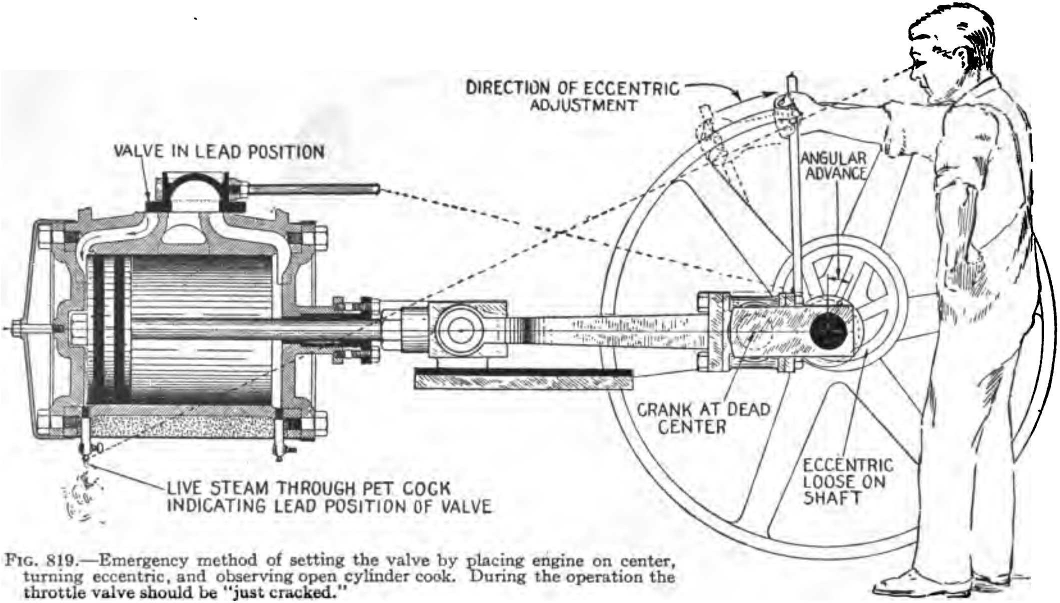
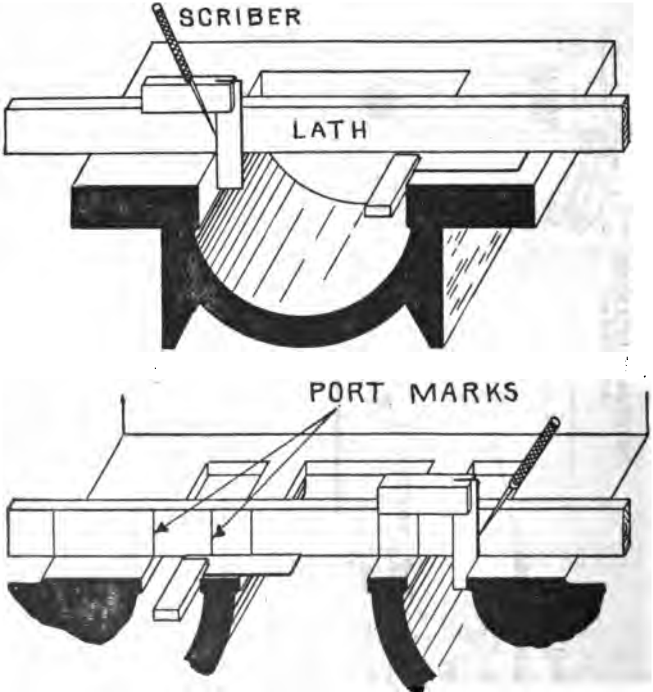

DIRECTION OF ECCENTRIC ADJUSTMENT
VALVE IN LEAD POSITION
ANGULAR ADVANCE
GRANK AT DEAD CENTER
ECCENTRIC LOOSE ON SHAFT
LIVE STEAM THROUGH PET COCK INDICATING LEAD POSITION OF VALVE

FIG. 819.—Emergency method of setting the valve by placing engine on center, turning eccentric, and observing open cylinder cook. During the operation the throttle valve should be "just cracked."

is admitted and the eccentric turned *forward* until steam issues from the cylinder cock, which indicates that the valve has lead, as in fig. 819. The eccentric is fastened in this position and the engine turned over to the other dead center. If steam escape from the cock at this end, the valve is in position for service until an opportunity occurs to open the steam chest in order to set the valve with more precision.

Taking Laths from the Valve and Seat.—It is desirable to have permanent records of the valve and seat dimensions, which are useful in setting the valve. These dimensions are transferred directly to laths or battens made of wood, and planed true and square; they should measure some three inches in width by five-eighths inch thick and of convenient length, depending on the size of the engine.



Figs. 820 and 821.—Preparing valve and seat laths. Care should be taken to place the laths square with the valve, or seat as indicated by use of the try square.

The valve is removed from the engine, and its steam and exhaust edges scribed on a lath as shown in fig. 820, care being taken to have the lath at right angles with the edges. Similarly the seat dimensions are transferred to a second lath as shown in fig. 821. The spaces on the laths representing the ports and exhaust cavity are painted white, and the valve faces and bridges black; the seat batten being painted black at the ends between the ports and the seat limits. The ends of the battens should be marked H and C, denoting "head end" and "crank end;" the completed battens appearing as in figs. 822 and 823.

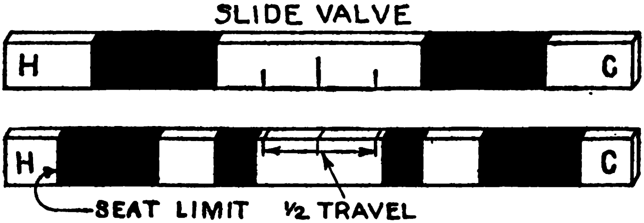


FIG. 822 and 823.—Slide valve battens. The ports and exhaust cavity are painted white, and the valve faces and bridges black. Lines are drawn corresponding to the half travel, or extreme positions, and the ends marked to distinguish head and crank end.

The batten is specially useful with engines having inside admission piston valves as a check on the valve setting.

The cylinder ends as well as the bushings, or valve seat should be painted on the batten as, in setting the valve, measurements are taken from the ends; the bushings are painted only part way across the batten to distinguish them from the cylinder ends, as shown in fig. 825. The travel of the valve is ascertained and indicated by lines which register as shown in the figures; thus the battens may be placed so as to show the extreme positions of the valve.

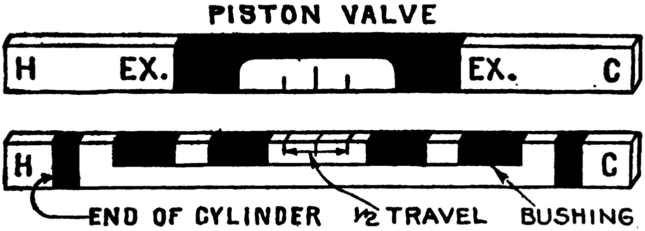


FIG. 824 and 825.—Inside admission piston valve battens. They are especially useful to check the valve setting. The ends of the cylinder should be indicated on the seat battens in addition to the bushings.

Ques. Of what particular use are battens?

Ans. They are helpful in the absence of an indicator to verify the setting of inside admission valves, and more especially to check the machine work on the valve and seat; if there be any errors in the location of ports, etc., they may be discovered by carefully transferring the various measurements of the valve and seat to battens for comparison.

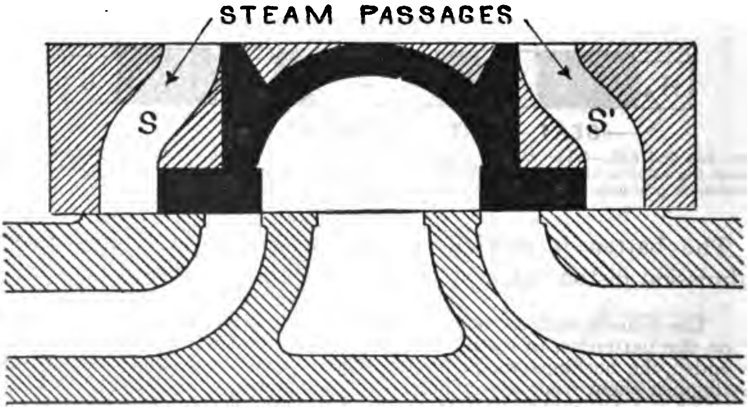


FIG. 826.—Showing similarity between the main valve of a riding cut off gear, and the ordinary slide valve. The main valve is simply a plain D valve, having steam passages S, S′ at its ends, and planed on its back to form a seat for the cut off valve.

Setting the Riding Cut Off Gear.—There are two types of riding cut off gear in general use, the first having a movable (rotating) eccentric, and the second a fixed eccentric, but having an adjustable cut off valve, known as the Meyer valve. The method of setting the valves for each type will now be described:

1. The Riding Cut Off, Movable Eccentric.

a. The main valve is set in the same manner as the ordinary slide valve.

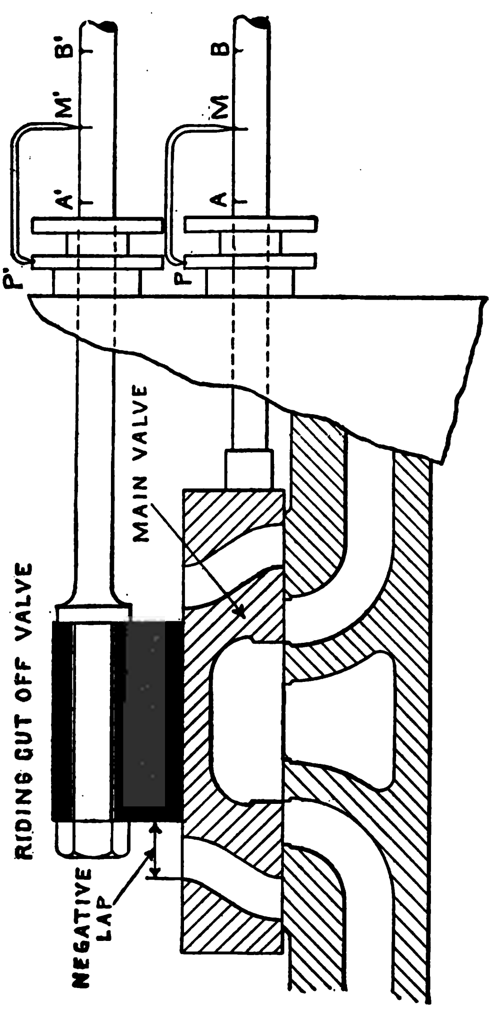


FIG. 827.—The riding cut off gear with both valves in neutral position. Tram marks M, M', are made on the stems to locate these positions, also A, B, and A', B', to locate the extreme positions. The same tram should be used for both valve stems.

To avoid confusion, it suffices to remember that the main valve is nothing more than an ordinary D valve having steam passages at its ends, and planed on its back to form a seat for the cut off valve. In setting the valve, therefore, the outer end walls are to be ignored.

The relation between the main valve, and an ordinary D valve is shown in fig. 826, the latter being illustrated in solid black section; the main valve has in addition to this, the portions shown at each end which contain the steam passages S and S'.

*b. The engine is now turned in the direction in which it is to run, until this valve is in its neutral position.**

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*NOTE.—The reason for putting the main valve in its neutral position is to facilitate this adjustment, as the necessary measurements are more conveniently made with respect to the main valve than to the seat.

To locate the neutral position, the engine is turned over until the main valve comes in the extreme positions A and B, as shown in fig. 827, and a reference mark for each, made on the valve stem with a tram, having one end in a convenient fixed center P. The distance A B, is bisected, giving the point M, the three points being permanently located with a center punch, and care being taken that the points are in a straight line parallel to the stem.

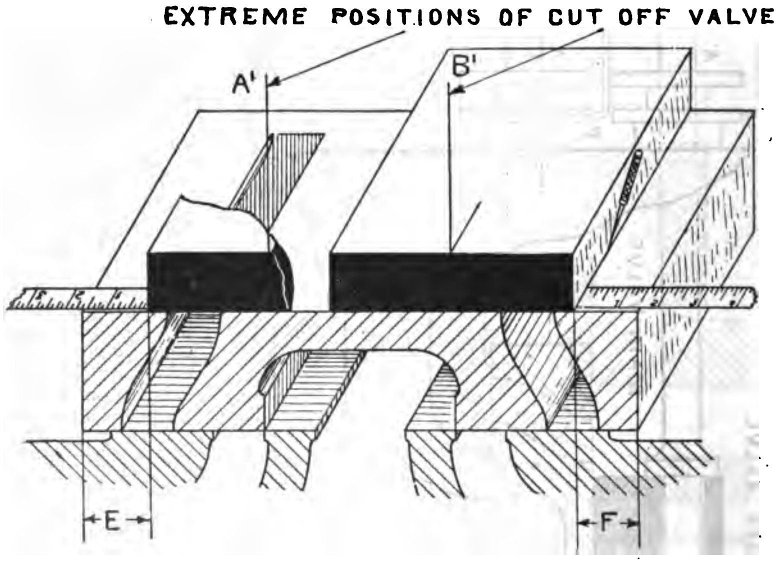

Fig 828.—Equalizing the travel of the riding valve. With the main valve in neutral position, and cut off eccentric loosened, the distances E and F, are measured for the two extreme positions and the valve stem adjusted until these distances become equal.

The valve is now placed in its neutral position, by turning the engine in the direction of its rotation until the point M, registers with the end of the tram as shown in the figure.

c. The next step is to *equalize the travel of the riding valve,* that is, *to adjust the riding valve stem or eccentric rod to the proper* *length so that the valve will travel an equal distance each side of its neutral position.*†

With the main valve in its neutral position, the movable eccentric is loosened on the shaft, and turned *in the direction in which the engine is to run* until the cut off valve is brought into its extreme positions A′ and B′, fig. 828.

The distance from the steam edge of the riding valve to the end of the main valve is measured for the two positions. If the valve stem be too long or too short these distances will be unequal; the valve stem or eccentric rod, in this case, should be adjusted until these distances E and F, are equal as shown in the figure.† Having equalized the travel of the valve, tram marks A′, B′, M′, indicating respectively the extreme and neutral positions, should be made as shown in fig. 827, on the cut off valve stem with the tram center at P′, using the same tram as was used for the main valve stem.

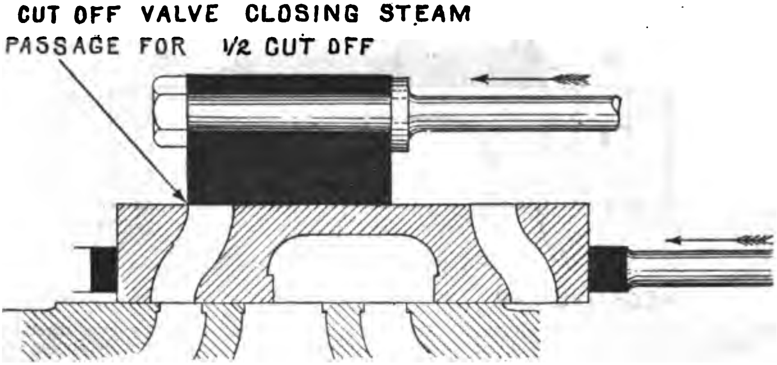

FIG. 829.—Riding valve in cut off position. To locate eccentric for any desired cut off, turn engine over till piston is at the desired point of cut off, then turn riding eccentric in the direction of motion till riding valve is in cut off position as shown.

d. To complete the setting, it remains only *to find the position of the movable eccentric which will give the desired cut off.*

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†NOTE.—To avoid error, it should be ascertained that the steam ports of the main valve are equidistant from the ends; if not, measurements E and F, fig. 828, should be taken with respect to the steam edges of the cut off valve, lines being lightly scribed on the back of the main valve to indicate the extreme positions.

If the valve gear is to cut off at, say one-half stroke, the engine is turned in the direction in which it is to run to this point of the stroke, and the movable eccentric turned in the same direction until the cut off valve has just closed the steam passage through the main valve as shown in fig. 413; the eccentric is now fastened in position.

e. When the riding cut off valve is operated by an automatic governor, as in many stationary engines, this last step is, of course, omitted.

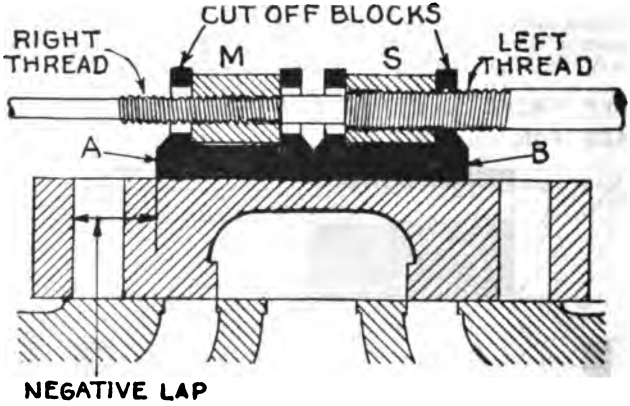


Fig. 830.—The riding cut off with fixed eccentric, showing cut off blocks M, S, screwed together when setting the valves. The cut off is latest when the blocks are together, depending on the negative lap. After setting the valves, the blocks should be fully extended for earliest cut off to see if there be any reopening after cut off and before the main valve has closed.

When the engine is provided with a governor, the travel of the cut off valve may in some cases be more conveniently equalized, by locating the center of the valve seat with a line scribed on the side or flange of the steam chest, and equalizing the travel of the cut off valve with respect to this line instead of the main valve. By this method it is not necessary to retain the main valve in its neutral position while adjusting the cut off valve; hence the loose eccentric need not be disconnected from the governor in making this adjustment.

2. The Riding Cut Off: Fixed Eccentric.

This type of riding cut off is set in much the same way as the preceding form. In making the adjustments, the important principle upon which the gear is based should be understood, and kept in mind, viz.: *the* **angular advance** *being fixed, the cut off is varied by changing the* **lap.** The valves are set as follows:

a. The main valve is set in the same manner as the ordinary slide valve.

b. To set the riding valve, the riding blocks M, S, *are first screwed closely together as shown in fig.* 830; *this being their position for* **latest** *cut off.*

c. The travel of the riding valve is now equalized by the method described on page 452, *and the riding eccentric located in the position best suited to the conditions under which the engine is to be operated.*

For a marine or reversing engine, the riding eccentric is set opposite the crank, that is, at 90 degrees angular advance, since the motion of the cut off valve is then correct for both forward and reverse motion; in this position an equivalent motion of the eccentric may be imparted to the valve stem by the cross head through a lever, thus dispensing with the eccentric. The angular advance of the riding eccentric on stationary engines is usually a little less than 90 degrees. The effect of reducing the angular advance is to require a smaller movement for a given change of cut off.

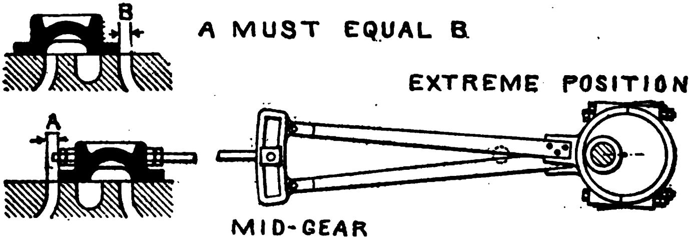
The riding eccentric should be so located that it will give the most rapid closure of the steam ports for the cut off mostly used.

d. The engine is now turned over to see if there be any reopening of the riding valve after it has cut off and before the main valve has cut off. Similarly it should be observed that there is no reopening for earliest cut off.

e. In case the cut off valve reopen before cut off of the main valve, this must be corrected by altering the position of the riding eccentric.

Setting a Link Motion.—Adjustments of the link gear should be such that the steam distribution will be favorable to smooth running and economy for the particular degree of expansion at which the engine is generally run. For instance, engines which work in full gear require a setting different from those which cut off short; the final adjustments therefore should be made with respect to obtaining the best results for the gear position mostly used. In general, setting a link motion comprises the following operations:

1. *Equalizing the travel;*
2. *Adjusting the eccentric rods to uniform length;*
3. *Finding the correct positions of the eccentrics;*
4. *Making final adjustments for best steam distribution in the gear position mostly used.*



FIGS. 831 and 832.—Setting the link motion: 1. *Equalizing the travel by adjusting the valve stem.* The eccentrics must be placed in extreme position and link in mid-gear.

Equalizing the Travel.—With the engine on the dead center, both eccentrics are turned on the shaft to the extreme position and the link placed in mid-gear as shown in figs. 831 and 832. If the length of the valve stem be correct, the valve should be in its extreme position, that is, the port opening A, for this dead center should be the same as port opening B, for the opposite center.

In case the port openings be unequal, the travel of the valve must be equalized by adjusting the length of the valve stem, or the eccentric rods, whichever be the more convenient.

Adjusting the Eccentric Rods to Uniform Length.—Both eccentric rods should be of the same length, and probably will be when not made adjustable. If both rods be of the same length, the position of the valve is unchanged for either full gear position *when the engine is on the dead center and the eccentrics are in the extreme position.*

To adjust the eccentric rods to uniform length, the engine is placed on the dead center, with the link in full gear and the eccentrics in the extreme position as shown in fig. 833.

The position of the valve or stem is marked in such a way that any movement in either direction can be measured. If both rods be of the same length, the mark on the valve stem will return to its original position when the link is shifted to its opposite or reverse full gear position; if the mark be displaced in either direction the reverse rod should be adjusted until the line returns to its original position. It is rarely necessary to make this adjustment, except when a gear has been completely dismantled.

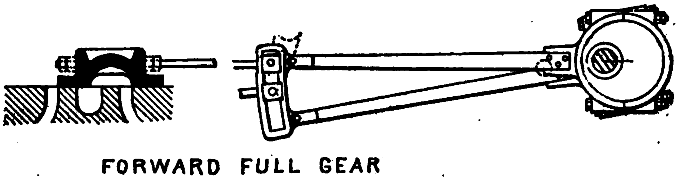


FIG. 833.—Setting the link motion: 2. *Adjusting the eccentric rods to uniform length.* The link is placed in forward full gear position. If rods be equal position of valve should be the same when link is shifted to reverse full gear.

Finding the Correct Positions for the Eccentrics.—With the engine on either center, the link is placed in the forward full gear position, and the eccentrics turned from the extreme position until they are at right angles with the crank, that is, where the angular advance equals zero as shown in fig. 834. The forward eccentric is now turned, *in the direction of forward rotation,* until the valve shows the desired lead as in fig. 835.

Similarly, the link is shifted to the reverse full gear position, fig. 836, and the reverse eccentric turned, *in the direction of reverse rotation,* until the proper lead is obtained as shown in fig. 837. Each eccentric is fastened after locating its position, and the results tested by trying the leads for the opposite centers, which completes the setting for engines operating in full gear.

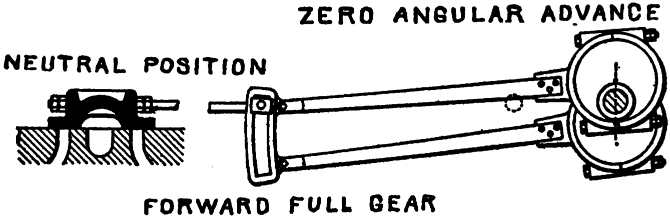


FIG. 835.—Setting the link motion: 3. *Locating the eccentrics—first step:* Both eccentrics are placed at zero angular advance, that is at right angles to the crank, bringing valve into neutral position.

Final Adjustments.—For engines which use the link motion as a variable expansion gear, the best results are obtained when the valve is set to give the proper lead for the "running cut off."

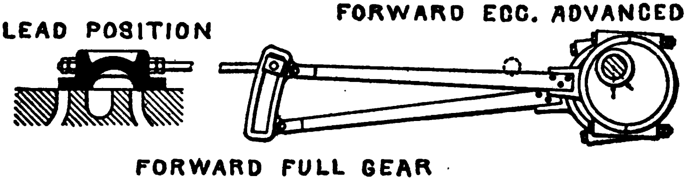


FIG. 834.—Setting the link motion: 3. *Locating the eccentrics—second step:* With link in forward full gear, the forward eccentric is advanced until valve shows proper lead, bringing valve into its forward linear advance position.

This applies especially to locomotives and engines which usually run with a considerable degree of expansion. Since the lead increases from full to mid-gear, it is obvious that if it be correct in full gear, it will be too great when hooked up for short cut off working.

The lead may be corrected for the running cut off, by placing the link in the running position, and setting back the forward or reverse eccentric, or both, in equal or unequal amounts until the desired lead is obtained.

The particular method of correcting the lead depends on the conditions of service. For engines which run mostly in forward gear, as express passenger engines, it is of little importance if both eccentrics have the same angular advance, however, for suburban tank locomotives, or those running considerable distances in each gear both eccentrics should have the same angular advance.

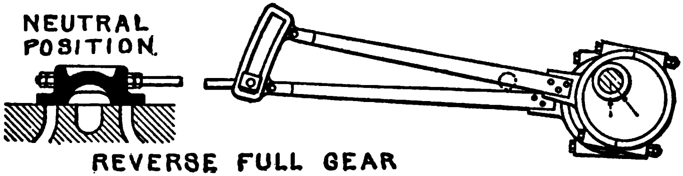


FIG. 836.—Setting the link motion: 4. *Locating the eccentrics*—***third step:*** The link is shifted to reverse full gear position, bringing valve back to its neutral position.

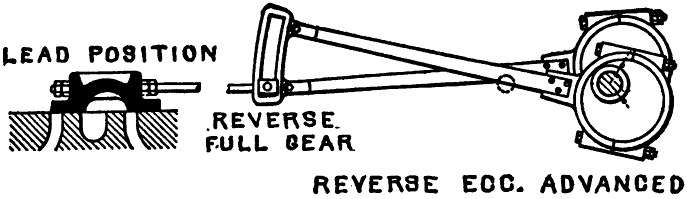


FIG. 837.—Setting the link motion: 4. *Locating the eccentrics*—***fourth step:*** The reverse eccentric is advanced until the valve shows the proper lead; this moves the valve to its reverse linear advance position which completes the setting except when final adjustments are made to adopt the link motion to special running conditions.

The following table shows the practice of several railroads with respect to the lead.

NOTE.—When the link block is at one end of the slot, the valve partakes of the motion of the eccentric attached to that end of the link. When the block is not at the end of the slot, the valve partakes of the combined motion of the two eccentrics, being the equivalent of ***a virtual eccentric** of decreased throw and increased angular advance.*

NOTE.—The object in curving a link block is to equalize the lead for all travels of the valve. To accomplish this it is necessary to have the radius of curvature of the slot such as will make the increase or decrease of the lead the same for both strokes of the piston.

Table of Leads for Locomotives

	Forward full gear	Reverse full gear	Lead for running cut off
Illinois Central	+1/32		+3/16
Chicago & Northwestern (Allen Valves)	−3/16		+1/4
New York, Hew Haven & Hartford	+1/16	−1/4	+1/4
Lake Shore & Michigan Southern	−1/16	−9/64	+5/16
Chicago Great Western	zero −3/64*	zero −3/64*	3/16 to 9/32 +1/4*

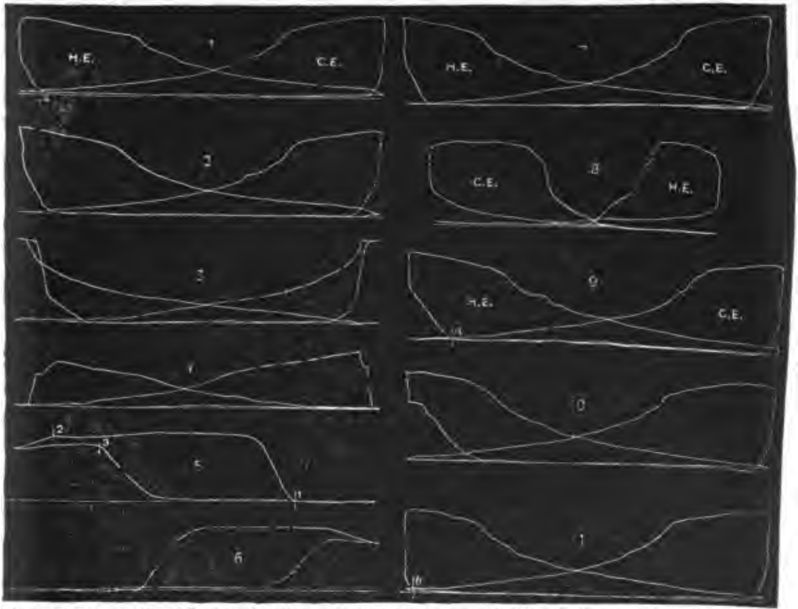
*For Mogul freight engines; with this exception the data in the table relates to passenger locomotives.

The diverse settings given in the table are due chiefly to the peculiarities of the several designs; the length of the eccentric rods has a marked influence on the methods followed in the different cases. It should be noted that the several railroads substantially agree on the amount of lead for the running cut off.

Valve Setting with the Indicator.—An important use of the indicator is to check the valve setting, for if the adjustments be not correct, the errors can be plainly located by taking cards. The accompanying cards show the distortions produced by various adjustments.

In setting a valve, changes in the steam distribution can be effected, either by shifting the position of the eccentric upon its shaft or by lengthening or shortening one or more of the rods connecting the eccentric with the valves. The result attained in either case should be clearly understood, since if it be attempted to make the needed adjustments in the wrong place, the engine may be put in worse condition than it was originally.

Changing the regular position of the eccentric either hastens or retards the action of the valves. Moving the eccentric ahead makes all the events of the stroke which are dependent upon the eccentric come earlier in the stroke, while moving back the eccentric causes all the events to occur later in the stroke.



FIGS. 838 to 848.—Effects of valve adjustments as recorded by indicator diagrams. Card No. 1 was taken when the eccentric had the proper angular advance of a little more than 90 degrees (in the case of a single eccentric Corliss engine) and cut off and compression were made as nearly equal as possible. In card No. 2 the angular advance was increased. In card No. 3 the angular advance was made still greater. In card No. 4 the eccentric was shifted back so that there was no angular advance. Card No. 5 is a crank end card and was taken when the eccentric was moved way back and gave a negative angular advance. The diagram was traced in the direction indicated by the numbers. Admission occurs at 1, cut off at 2, release at 3. Card No. 6 is a similar diagram from the head end. In card No. 7 the eccentric is moved back and the rods adjusted to give correct steam distribution and then under these conditions the eccentric card, No. 8, was taken. In this case the indicator derived its motion from the eccentric rod instead of from the cross head and the diagram shows the working of the valves of the engine, when the indicator drum is moving at its greatest speed; and thus any peculiarities in the action of the valve are magnified. Compression and release come near the middle of the card and are spread over a considerable length. In card No. 9 the head end exhaust rod was lengthened and there is too much compression at *a*. In No. 10 the rod is made still longer. In card No. 11 the head end exhaust rod is made too short and there is too little compression at *b*.

NOTE.—***When setting an eccentric,*** a rule that can be easily remembered is: *Set the eccentric far enough ahead of a right angle to the crank to allow for the lap and lead of the valve.* The mistake of turning the eccentric "just half way around" to reverse the engine should not be made.

NOTE.—***An engine properly built,*** and not run at too high a rotative speed, will run smoothly with a moderate amount of compression. To attempt to get smooth running with an extra amount of compression or lead means more oil, more coal, and more repairs.

In the Corliss engine the point of cut off is determined by the governor instead of by the eccentric, and so only the points of admission, release and compression are affected by shifting the eccentric. In the case of a slide valve engine it affects all the events.

In a slide valve engine an adjustment of the length of the eccentric or valve rod simply changes the position of the valve so that it will have more lap or lead at one end than before, and less lap or lead, as the case may be, at the other end.

In the Corliss engine an adjustment of the eccentric rod produces the same result, increasing or decreasing the lap or lead, as the case may be, at one end of the stroke, while it has the opposite effect at the other end.

The lengths of the eccentric rod or gab rod on a Corliss engine, however, should never be changed, unless it is found that the intermediate rocker and wrist plate do not travel equally on each side of a vertical center line.

All the rod adjustment should be made in the radial rods extending from the wrist plate to the four valves. Adjusting any of these rods, of course, affects only the valve to which each rod is connected and will give greater or less lap to that valve, according as it is lengthened or shortened.

If more lap were given to an exhaust valve in this way, for example, the valve would open later and compression would occur earlier, since the valve would close earlier. The port opening would also be less. If less lap were given to the valve the reverse of these conditions would be true.

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